CONVERSION OF AN EXISTING GAS TURBINE TO AN INTERCOOLED EXHAUST-HEATED COAL-BURNING ENGINE

by

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ABSTRACT

An existing gas-turbine engine has been selected and modified "on paper" to accommodate an innovative, high-efficiency thermodynamic cycle. The modified Solar 5650 industrial gas turbine burns coal in an intercooled exhaust-heated cycle for power generation. This thesis focuses on the alterations that must be made to this off-the-shelf engine and their impact on the overall performance of the engine.

The conversion process involves optimizing the exhaust-heated cycle to obtain peak thermal efficiency and near-maximum specific power. Three design changes are explored to optimize the intercooled exhaust-heated 5650 cycle. The alternatives include running the intercooled exhaust-heated 5650 at a slower speed with no turbomachinery modifications, running the engine at its design pressure ratio, or redesigning all of the turbomachinery. Each of these options and a cycle modification, increased turbine-inlet temperature, are measured on performance and life-cycle-cost bases. Sizing analysis for a rotary regenerator heat exchanger and combustor recommendations for the cycle are also included.

The results of this study indicate that the performance benefit gained by redesigning the turbomachinery outweighs its extra initial capital cost. The other options analyzed are more expensive to operate than the base 5650 unit. The increased turbine-inlet temperature modification resulted in better performance and cost than any of the options. Running the converted engined at its original design pressure ratio was also considerably attractive due to its lower capital costs.

This thesis is one part of a three-part project sponsored by the U. S. Department of Energy and supervised by MIT Professor David Gordon Wilson. The other two parts are the preliminary design of an optimal or "blue-sky" exhaust-heated, coal-burning engine and the cold coal-ash-fouling test of a rotary regenerator.

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As always, I am thankful to my family for their prayerful support and encouragement throughout my life. I must also acknowledge The Lord Jesus Christ who has ordered all of this and who has blessed me beyond all my expectations.

Unless the Lord builds the house,
They labor in vain who build it;
Unless the Lord guards the city,
The watchman keeps awake in vain.
It is vain for you to rise up early
To retire late,
To eat the bread of painful labors;
For He gives to His beloved even in his sleep.

TABLE OF CONTENTS

		PAGE
ABSTRACT		2
ACKNOWLEDGEMENTS		3
TABLE OF CONTENTS		4
LIST OF FIGURES		6
LIST OF TABLES		8
NOMENCLATURE		9
1.0	INTRODUCTION	10
1.1	Objective	10
1.2	Background	11
1.3	Direct-Fired Units	12
1.4	Indirect-Fired Units	14
1.5	Indirect-Fired Exhaust-Heated Units	15
2.0	GAS-TURBINE CYCLES	19
2.1	The Simple Cycle	19
2.2	The Recuperated Cycle	22
2.3	The Intercooled Recuperated	24
2.4	The Intercooled Exhaust-Heated Cycle	26
3.0	APPLICABLE TECHNOLOGIES AND DESIGN PHILOSOPHY	29
3.1	High-Efficiency Complex Systems	29
3.2	Recuperation	30
3.3	Intercooling	33
3.4	Combining Intercooling and Recuperation	35
3.5	Optimum Pressure Ratios	36
3.6	Variable-Area Power-Turbine Nozzles	38
4.0	ENGINE SELECTION	41
4.1	The Selection Process	41
4.2	The Solar 5650 Features	42
4.3	Solar 5650 Performance	43
5.0	ENGINE CONVERSION	46
5.1	Analysis Overview	46
5.2	Base Intercooled Solar 5650 Performance Data-Match	47
5.3	Intercooled Exhaust-Heated Solar 5650 Cycle Performance Prediction	50
5.4	Optimal Intercooled Exhaust-Heated 5650 Cycle Pressure Ratio	57
6.0	THE INTERCOOLED DESIGN OPTIONS	61
6.1	Applicable Nomenclature and Base Engine Data	61
6.2	Option 1 - Run Engine at Original Pressure Ratio	62
6.3	Option 2 - Redesign All Turbomachinery	63
6.4	Option 3 - Run Existing Turbomachinery Off-Design	74
7.0	INCREASED TURBINE-INLET TEMPERATURE	82
8.0	REGENERATOR-MATRIX SIZING EFFECTS	84
9.0	ECONOMIC COST-BENEFIT ANALYSIS	89
9.1	The Life-Cycle-Cost Method	89
9.2	The Life-Cycle Equation Unknowns	90
9.3	Blue Sky and Optimal Off-the-Shelf Design Comparison	90
10.0	COMBUSTOR RECOMMENDATIONS	98
11.0	CONCLUSIONS AND RECOMMENDATIONS	100

REFERENCES 101
APPENDIX 1 Intercooled Solar 5650 Cycle Computer Model 104
APPENDIX 2 Intercooled Exhaust-Heated Cycle Computer Model 111
APPENDIX 3 Off-Design Performance Computer Program 124

LIST OF FIGURES

		PAGE
Figure 1.1	DOE/METC Sponsored System Descriptions	12
Figure 1.2	Closed-Cycle Coal-Burning Gas Turbine	15
Figure 1.3	Exhaust-Heated Coal-burning Gas Turbine	16
Figure 1.4	Mordell's Exhaust-Heated Gas Turbine	17
Figure 1.5	Rotating Ceramic-Matrix Regenerator	18
Figure 2.1	Thermal Efficiency vs. Specific Power - CBE	20
Figure 2.2	CBE Schematic and T-s Diagram	21
Figure 2.3	CBEX Schematic and T-s Diagram	23
Figure 2.4	Thermal Efficiency vs. Specific Power - CBEX	24
Figure 2.5	CICBEX Schematic and T-s Diagram	25
Figure 2.6	Thermal Efficiency vs. Specific Power - CICBEX	26
Figure 2.7	CICXEB Schematic and T-s Diagram	27
Figure 2.8	Rotary Regenerator	27
Figure 3.1	Effect of Thermodynamic Cycle Selection	30
Figure 3.2	Brayton Cycle - Effect of Recuperation	32
Figure 3.3	Recuperated and Simple Cycle Comparison	33
Figure 3.4	Brayton Cycle - Effect of Intercooling	34
Figure 3.5	Recuperated Brayton Cycle - Effect of Intercooling	37
Figure 3.6	Nozzle-Vane Positions	40
Figure 3.7	Effect of Variable Power-Turbine Stators	40
Figure 4.1	Solar 5650 Industrial Gas Turbine	43
Figure 4.2	Solar 5650 Engine Cross-Section	44
Figure 5.1	Intercooled Exhaust-Heated 5650	56
Figure 5.2	Design-Point Thermal Efficiency vs. Specific Power	58
Figure 6.1	Compressor Velocity-Diagram Conventions	61
Figure 6.2	Turbine Velocity-Diagram Conventions	62
Figure 6.3	Stage 1 Schematic	67
Figure 6.4	Stage 2 Schematic	68
Figure 6.5	Baseline 5650 Turbine Velocity-Diagrams	70
Figure 6.6	Velocity-Diagrams for Redesigned Turbines	71
Figure 6.7	Turbine Schematic	73
Figure 6.8	Solar 5650 Compressor First Stage Operating Line	75
Figure 6.9	Solar 5650 Compressor Second Stage Operating Line	76
Figure 6.10	Solar 5650 Turbine Operating Line	77
Figure 6.11	Thermal Efficiency Comparison	79
Figure 6.12	Off-Design Thermal Efficiency vs. Specific Power	80
Figure 7.1	Composite Performance	83
Figure 8.1	Regenerator Disc Mass vs. Hydraulic Diameter	85
Figure 8.2	Regenerator Disc Diameter vs. Hydraulic Diameter	86
Figure 8.3	Regenerator Disc Thickness vs. Hydraulic Diameter	87
Figure 8.4	Cycle Output Power vs. Hydraulic Diameter	88

Figure 9.1 Life-Cycle Cost..95
Figure 9.2 Relative Life-Cycle Cost Composite Charts.................................96

LIST OF TABLES

		PAGE
Table 4.1	Candidate Gas Turbines	42
Table 4.2	Engine Dimensions	44
Table 4.3	Base Solar 5650 Design-Point Performance	45
Table 5.1	Intercooled Base 5650 Design-Point Cycle Parameters	48
Table 5.2	Overall Intercooled Performance: Model vs. Engine Data	49
Table 5.3	Component Performance: Intercooled Model vs. Engine Data	50
Table 5.4	Ceramic Matrix Surface Geometry	51
Table 5.5	Regenerator Dimensions and Performance	52
Table 5.6	Coal Ultimate Analysis	53
Table 5.7	Overall Performance Comparison: IC Base vs. IC Exhaust-Heated	53
Table 5.8	Component Performance: IC Base vs. IC Exhaust-Heated	54
Table 5.9	Overall Performance Comparison: IC Exhaust-Heated vs. Optimal	59
Table 5.10	Component Performance Comparison: Exhaust-Heated vs. Optimal	59
Table 5.11	Comparison of Regenerator Dimensions and Performance	60
Table 6.1	Base 5650 Compressor Velocity-Diagram Data	62
Table 6.2	Base 5650 Compressor Dimensions	62
Table 6.3	Overall Performance Comparison: IC Base vs. IC Exhaust-Heated	63
Table 6.4	Centrifugal-Compressor Design Assumptions	65
Table 6.5	Intercooled Compressor Design Parameters	66
Table 6.6	Comparison of Redesigned Compressor Geometry	66
Table 6.7	Redesigned Compressor Velocity-Diagram Data	67
Table 6.8	Intercooled Compressor Design Intent vs. Prediction	68
Table 6.9	Mean-Diameter Turbine Velocity-Diagram Data	69
Table 6.10	Gas-Producer Turbine Geometry	69
Table 6.11	Power-Turbine Geometry	72
Table 6.12	Mean-Diameter Turbine Velocity-Diagram Data	72
Table 6.13	Design Intent vs. Predicted Efficiency	73
Table 6.14	Overall Performance Comparison: Optimal vs. Off-Design	78
Table 6.15	Overall Performance Comparison: Optimal vs. Off-Design	81
Table 7.1	1339 T.I.T. Cycle Comparison	82
Table 8.1	Surface Geometry for Three Cores	84
Table 8.2	Core Run Data	88
Table 9.1	Life-Cycle Calculation Constants	90
Table 9.2	Initial Production Costs	92
Table 9.3	Life-Cycle Cost Variables	93
Table 9.4	Life-Cycle Cost Summary	93
Table 9.5	Optimal Design Life-cycle Comparison	95
Table 9.6	Comparison of Life-Cycle Cost Variables	96
Table 9.7	CICXEB and ICEH Redesign Component Comparison	96

NOMENCLATURE

A	area	(m^2)
b	impeller width	(mm)
C	absolute velocity	(m/s)
d	diameter	(mm)
MBTU	million-Btu	
P	pressure	(kPa)
r	compressor pressure ratio	
Rn	reaction	
s	entropy	(kJ/kg K)
T	temperature	(K)
T'	turbine-inlet-to compressor-inlet temperature ratio	
u	rotor velocity	(m/s)
W	relative velocity	(m/s)
WA1	compressor-inlet airflow	(kg/s)
Z	number of blades	

<u>Greek Letters</u>

α	fluid angle	(°)
β	blade angle	(°)
Δ	difference operator	
η	efficiency	
λ	blade hub-to-tip ratio	
ϕ	flow coefficient	
ψ	blade loading coefficient	

<u>Subscripts</u>

1	inlet (impeller or blade row)
2	exit (impeller or blade row)
3	vaneless difuser exit
4	vaned diffuser exit
c	absolute
h	hub
n	annulus
s	shroud
t	tip
th	thermal
w	relative

1.0 INTRODUCTION

This chapter presents a description of the topic objective and background including an overview of previous research as well as a brief historical perspective. Proposed cycles for coal burning in gas turbines are presented with a final discussion on the advantages of the exhaust-heated cycle.

1.1 Objective

This thesis focuses on the modifications necessary to convert an existing gas turbine to an intercooled exhaust-heated, coal-burning engine and the resulting performance of the modified engine. Although coal has been selected as the primary fuel for consideration, a section on the possibilities of using biomass is also included. The engine chosen for conversion is the 2.8 MW Solar 5650 industrial gas turbine. The conversion process involves optimizing the intercooled exhaust-heated cycle to obtain peak thermal efficiency and near-maximum specific power through the consideration of three design changes. The alternatives include both running the intercooled exhaust-heated 5650 at design speed and at a slower speed with no turbomachinery modifications, or redesigning all of the turbomachinery. Each of these options and two cycle modifications are examined on a life-cycle-cost basis. Previously developed methods of heat-exchanger sizing, centrifugal-compressor design, turbine design, and performance prediction are used extensively to arrive at the final results. The reader is encouraged to refer to cited references when detailed explanations are desired.

This thesis is one part of a three-part project sponsored by the U. S. Department of Energy (DOE contract # DE-AC21-89MC26051) and supervised by MIT Professor David Gordon Wilson. The other two parts are the preliminary design of an optimal or "blue-sky" exhaust-heated, coal-burning engine, and the cold coal-ash-fouling test of a rotary regenerator.

1.2 Background

Studies have been conducted since the 1930s to develop feasible coal-burning gas turbines with the first prototypes used in German locomotives. After the introduction of the first aircraft gas turbines, a project was started by the Locomotive Development Committee (a consortium of six railroad and six coal companies) in 1944 to develop a coal-burning gas turbine within the United States. The project concluded after a 1000-hour endurance test revealed catastrophic turbine erosion [1]. The U.S. Bureau of mines and the Australian Aeronautical Research Laboratories also attempted separate experiments in burning uncleaned, unprocessed coal in gas turbines that ended with failure [2].

A more recent study conducted in 1982 by General Motors involved using a direct-fired coal-burning gas turbine as a prime mover for a Cadillac Eldorado. This project was moderately successful as the coal had been pulverized to an average size of 53 microns and cleaned of ash and sulfur. Resulting thermal efficiency of the recuperated gas turbine was quite favorable [2].

1982 is also a significant year because the increasing price gap between coal and other forms of fossil fuels as well as projections of depleting petroleum resources prompted the Department of Energy (DOE) to begin research in coal-burning heat engines [3]. Both research in diesel and gas turbine engines was funded. There would be great advantages to the development of new forms of coal-fired propulsion and power generation systems which incorporate appropriate stringent pollution controls. Oil had accounted for 42 percent of the fuel consumed in the U.S. in 1988 and domestic oil production was at its lowest point in 25 years for the first half of 1989 [4]. Also, the political instability of the middle east and recent invasion of Kuwait by Iraq demands that conservation and the use of other fuels must become a balanced portion of the U.S. strategy to decrease dependence on imported oil.

The gas turbine has the advantages of compact size, potential low cost, and relative ease of control over the Rankine and Diesel cycles which tend towards larger size and increased

acquisition cost. These reasons combined with the abundance of coal reserves has made the prospect of coal -fired gas turbines extremely attractive.

1.3 Direct-Fired Units

DOE has awarded contracts to General Electric (GE), Westinghouse, Allison Gas Turbines, and Solar Turbines for the development of integrated coal-fired gas turbine systems. These four corporations have concentrated their efforts on direct-fired units as summarized in figure 1.1. Direct-fired units have combustion of the compressed air with coal prior to entering the turbine. This cycle is essentially the simple gas turbine cycle and is illustrated in figure 2.2. The air is first compressed in a compressor and then flows through a slagging coal combustor which usually performs some type of hot-gas cleanup. Products of combustion enter the expander or turbine directly, perform work, and are rejected to a sink which may be the atmosphere or waste heat recovery system. Further pollutant removal is necessary prior to leaving the stack.

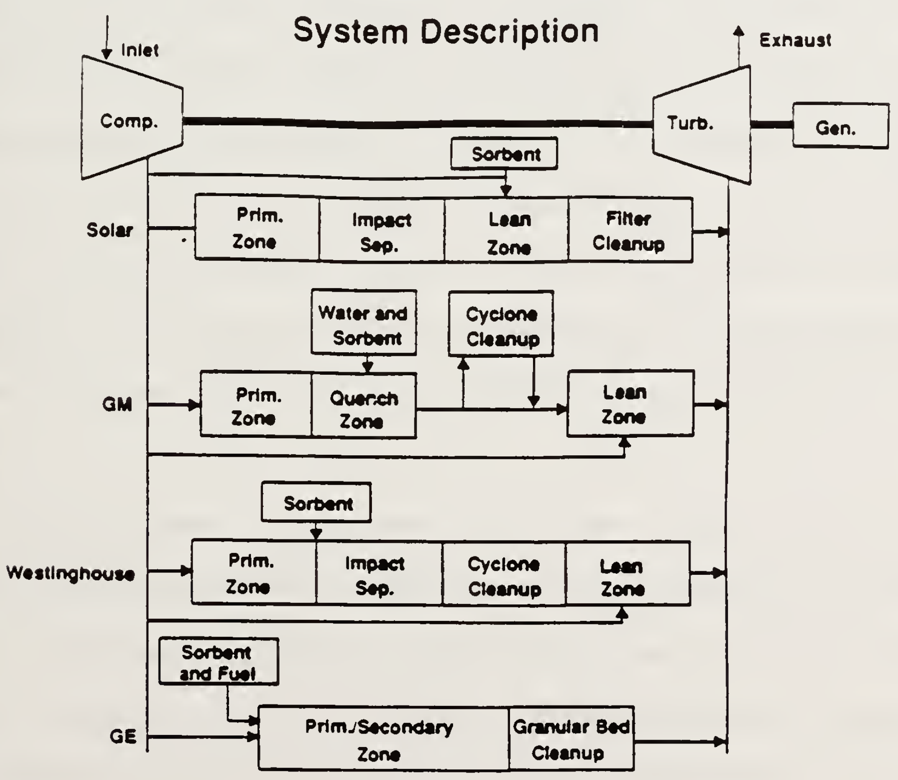


Figure 1.1 DOE/METC Sponsored System Descriptions [3]

Variations of the direct-fired coal-burning cycle are similar to those incorporated in industrial gas-turbine engines to increase specific power, thermal efficiency or part-load performance. These changes include the addition of recuperators or intercoolers. Changes may also be made to the combustor and waste heat may be linked to a separate steam cycle in order to utilize excess heat produced in the combustor [5].

The direct-fired units face problems, both technical and economic in nature. The coal combustion process always results in ash and alkali-laden gas which can result in particulate and chemical action on the turbine as well as pollution. Particulate matter has a powerful erosive effect on the turbine blades and even if reduced to an acceptable size (5 microns for gas turbines), alkalinity of the combustion products still poses a problem [6]. Over periods of time, ash also tends to form deposits around the blades that have deleterious aerodynamic effects on their performance [1,7].

The combustion process required for direct-fired units also increases in complexity because it involves feeding and burning coal at higher than atmospheric pressure. Past experience of two reputable companies conducting research in this area has indicated that uniform injection of dry micronized coal (DMC) into a pressurized combustor is a problem not easily solved [3]. The Avco Research Laboratory/Textron is the only sponsored facility which is currently advocating a slagging combustor which utilizes DMC pressurized to 6 atmospheres. Avco chose this combustion system based on the increased treatment costs of using coal-water slurry (CWS) as the other DOE-sponsored activities have advocated [6].

Solutions to some of the problems encountered by direct-fired coal-burning gas turbines include: implementing various types of hot-gas cleanup, varying blade alloys to gain the needed erosion resistance, designing appropriate aerodynamic blade profiles to minimize the effects of solids in the airstream, experimenting with various sorbents within the combustor to reduce the deposition rate of ash, and maintaining low blade-surface temperatures to inhibit ash stickiness and agglomeration [8,9,10,11]. These proposed

solutions do not always produce the predicted results due to an incomplete understanding of coal ash deposition. Australian researchers utilizing native brown coal found that when larger ash particles were removed with cyclone separators, ash deposition rates actually rose in the blades by 70 percent [12]. Also, studies at DOE METC found that when ash deposition was reduced the remaining deposited materials adhered much more strongly to metal surfaces. The nature of coal-ash deposition must first be better understood before direct-fired units become feasible and reliable enough for commercial use.

1.4 Indirect-Fired Units

The cycle which this thesis investigates and which currently receives less attention is the indirect-fired gas turbine. This type of cycle is composed of both closed and exhaust-heated cycles [8,13].

The indirect-fired closed-cycle gas turbine operates with the working fluid completely separated from the products of combustion. Energy is transferred to the expander via some type of highly effective heat exchanger. Figure 1.2 depicts the closed-cycle gas turbine. The closed cycle avoids the problems of ash deposition associated with direct-fired cycles. The working fluid is not restricted to air and may be pressurized which results in compact engine components [14]. thermal efficiencies to 55 percent have been predicted but operating units have attained efficiencies between 28 and 30 percent [15].

Because of the problems associated with ash deposition and fouling, closed-cycle engines are presently the only available coal-fired gas turbines. These units have excellent part-power efficiencies but design efficiency is dependent upon the heat transfer between the high-pressure gas and the heat exchanger wall. Maximum temperatures are limited by the working fluid's maximum temperature which is constrained by the present state of technology to about 1100 K (1600 F). The increased complexity of these cycles as well as a large additional heat-exchanger and gas cooler make these engines less economically attractive due to high initial capital investment.

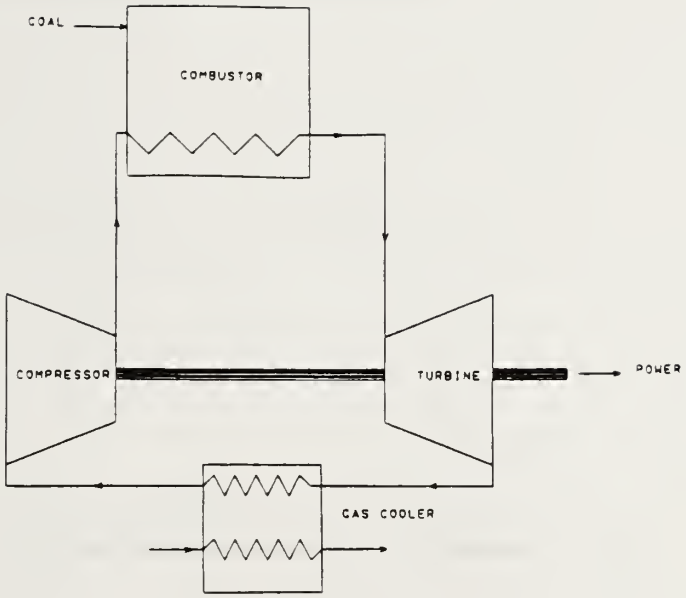


Figure 1.2 Closed-Cycle Coal-Burning Gas Turbine [16]

1.5 Indirect-Fired Exhaust-Heated Units

The exhaust-heated cycle involves transposing the combustor from its position after the compressor in the simple direct-fired gas turbine to a location after the turbine. The entire heat addition to the air entering the expander occurs through a heat exchanger. The cycle is illustrated in figure 1.3.

The exhaust-heated cycle was first studied extensively from 1949 through 1957 by Professor D. L. Mordell of McGill University. His preliminary analysis concluded that a heat-exchanger effectiveness of at least 75 percent was necessary to make this cycle attractive. The exhaust-heated cycle will yield the same specific power as a conventional open-cycle gas turbine with the same temperature ratio, pressure ratio, and component

efficiencies. Thermal efficiencies for both cycles would be equivalent if the heat-exchanger effectiveness for the exhaust-heated cycle were 100 percent.

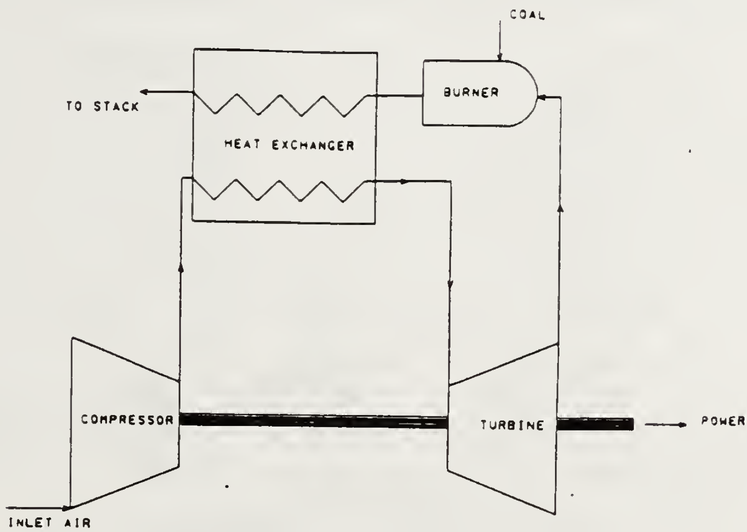


Figure 1.3 Exhaust-Heated Coal-Burning Gas Turbine [16]

Mordell's initial exhaust-heated test rig (figure 1.4) was constructed of a Rolls Royce Dart gas turbine, a unique shell-and-tube heat-exchanger, and a slagging cyclone combustor. He used a screw-type coal feeder to provide fuel-feed uniformity since the combustor was operating at atmospheric pressure. The slagging combustor was well suited for atmospheric conditions and for the high turbine-exit air temperatures. Although dry-ash fouling of the heat-exchanger surfaces was not as critical as he first predicted, there were some clogging problems associated with the combustor, large pressure losses in the heat-exchanger, and corrosion due to sulfur condensation in the heat-exchanger tubes. Mordell's experiments demonstrated that the exhaust-heated cycle is a feasible method of burning coal in a gas turbine if heat-exchanger fouling can be limited and effectiveness optimized [12].

The advantages of this cycle combine those of both direct-fired and closed cycle units. The products of combustion never pass through the turbine, air is used as the working fluid, and the combustor operates at atmospheric pressure. The major concern regarding ash deposition, erosion, and corrosion of the turbine blades is alleviated. The critical

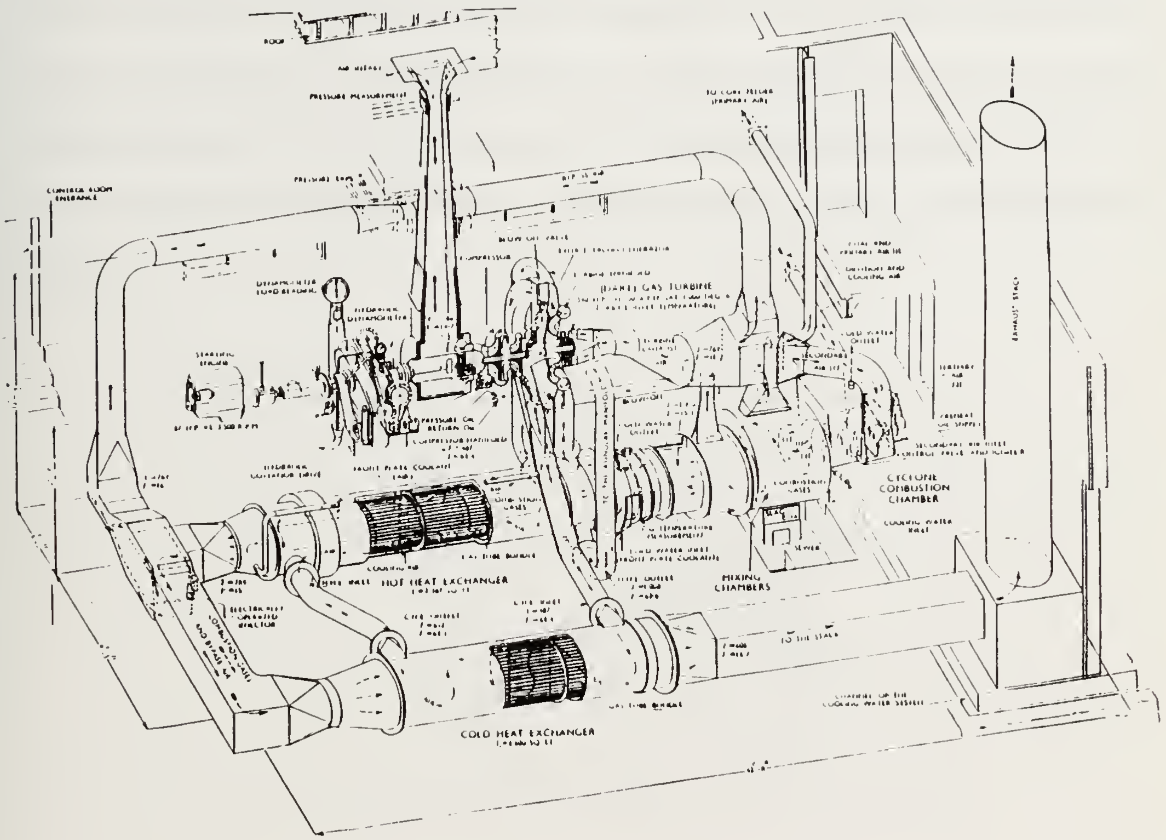


Figure 1.4 Mordell's Exhaust-Heated Gas Turbine [12]

component now becomes the heat-exchanger rather than the turbine. Choice of a proper heat exchanger should take into account capital cost as periodic replacement or cleaning will be necessary. For this study, involving the conversion of a commercially available engine, the rotating ceramic matrix was chosen for the exhaust-heated cycle. In the ceramic regenerator shown in figure 1.5, the two streams, one of compressed air and the other containing products of combustion, pass through the annular area of the matrix in counterflow. The disk rotates and the matrix absorbs heat from the hot stream and transfers it to the cold stream. This feature of rotation provides an interesting benefit in that

the matrix tends to be "self cleaning." As the matrix rotates, flow direction between the hot and cold sides reverses and any dry deposits which may have formed as the hot exhaust gases pass through in one direction should be dislodged when the compressed air from the compressor flows in the opposite direction. Circumferential and radial seal on the surface of the matrix prevent the streams of gas from mixing. The ceramic matrix was also chosen because of its suitability for high temperatures in a low-pressure ratio cycle [17]. This type of heat-exchanger has an effectiveness of over 0.95 as used in the Allison GT 404. An effectiveness of 0.975 could be obtained on this engine if the current matrix thickness were doubled.

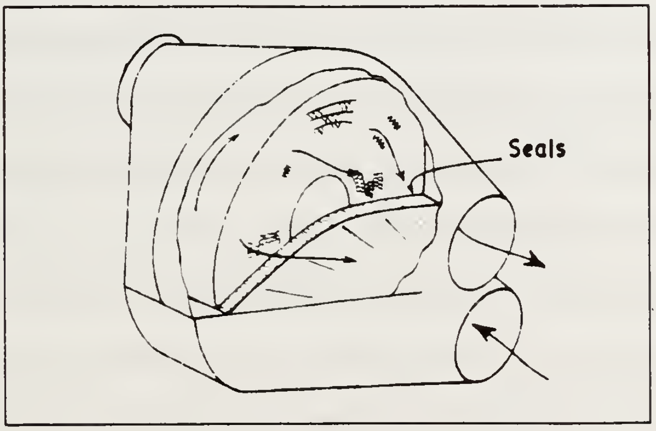


Figure 1.5 Rotating Ceramic-Matrix Regenerator [5]

2.0 GAS-TURBINE CYCLES

This section discusses various types of gas turbine cycles and compares their overall performance using specific power and thermal efficiency as primary parameters. Specific power is the power output of the cycle normalized by the product of the mass flow rate, specific-heat capacity and stagnation temperature at inlet. Thermal efficiency is defined as the net power output of the cycle divided by the rate of energy addition during the combustion process. These parameters will be used to explain performance comparisons throughout this section. The advantages and reasons for choosing the intercooled exhaust-heated cycle for this study will then be apparent.

2.1 The Simple Cycle

Most existing gas turbines use a simple direct-fired cycle operating on a well refined grade of petroleum based fuel. This simple cycle is independent on increasing turbine inlet temperature (TIT) and pressure ratio in order to attain higher thermal efficiency and specific power as shown in figure 2.1 [5]. This cycle is illustrated schematically in figure 2.2 and is composed of a compressor, combustor and expander. Following the guidelines in Wilson [5], the cycle can be referred to as a Compressor-Burner-Expander (CBE) cycle. In this type of nomenclature the symbols represent the following components:

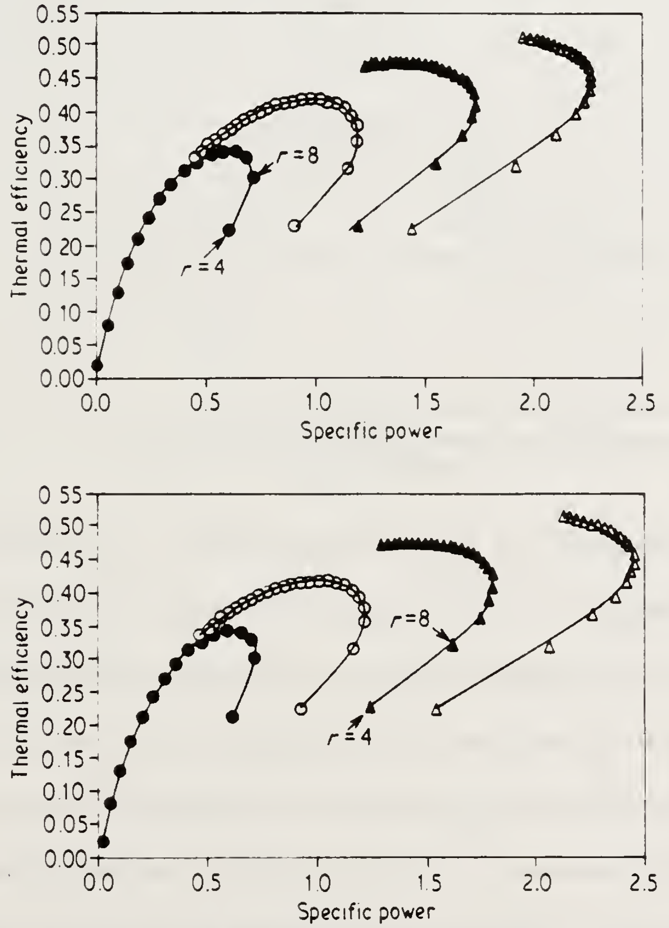
C ≡ Compressor
B ≡ Heat addition from an external source (i.e. combustor or burner)
E ≡ Expander (i.e. turbine or exhaust nozzle)

and appear in the order in which the components they represent are encountered by the working fluid. In addition, the symbols

I ≡ Intercooler
X ≡ Exhaust-gas-to-compressed-air heat exchanger

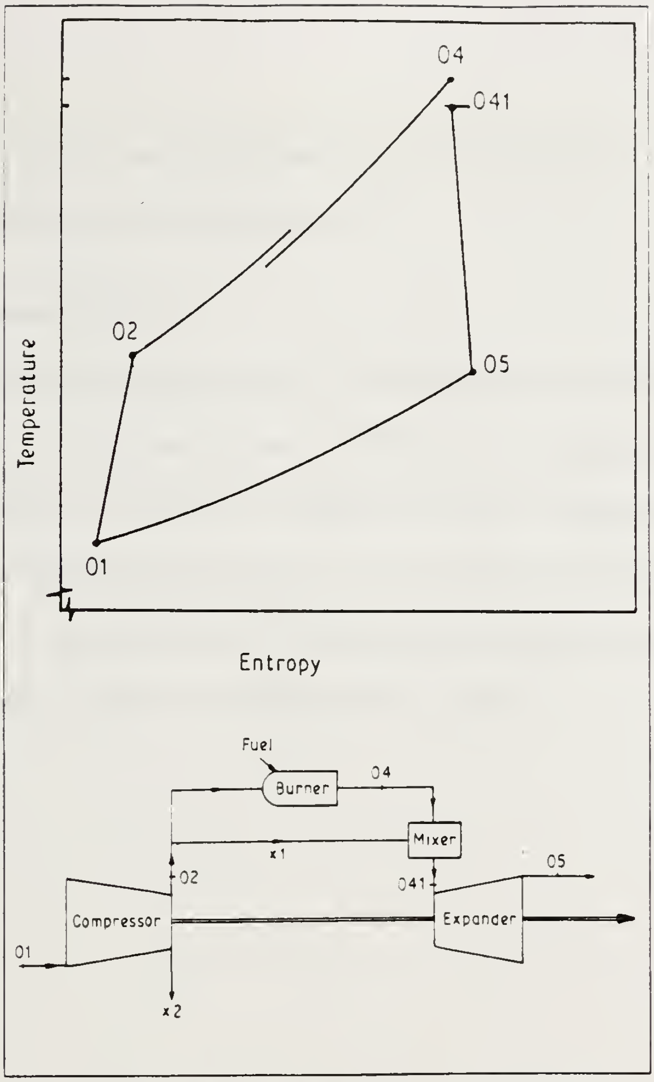
will be needed later. The symbol X is used only in the expander-exhaust position even though the working fluid passes through the heat exchanger twice.

A temperature-entropy (T-s) diagram in figure 2.2 illustrates the various component contributions to the simple-cycle. The pressure and temperature of the working fluid are increased in the compressor (01-02). External heat is added at a relatively constant pressure in the combustor (02-04). The turbine or expander extracts work from the high-temperature-and-pressure gas (041-05). Energy in excess of that needed to drive the compressor is then used for power generation or propulsion depending on the duty of the turbine.



Plot of simple cycle thermal efficiency against specific power for large (upper) and small (lower) cooling-mass-flow fraction. ●, T′ = 4; ○, T′ = 5; ▲, T′ = 6; △, T′ = 7

Figure 2.1 Thermal Efficiency vs. Specific Power - CBE [18]



Plot of temperature against entropy and block diagram for simple cycle

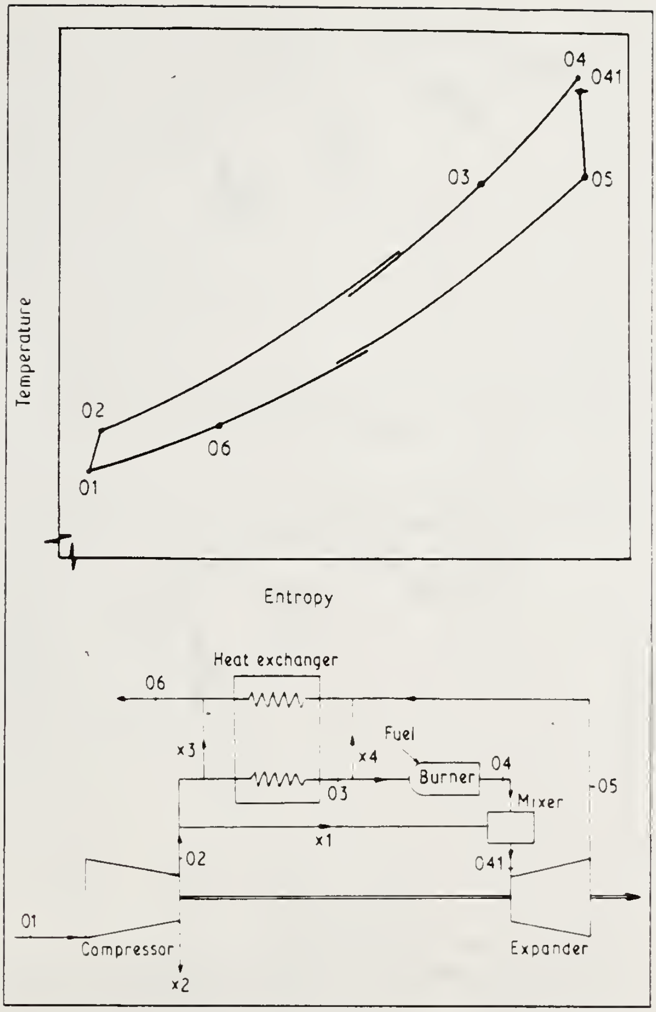
Figure 2.2 CBE Schematic and T-s Diagram [18]

In the simple cycle with a fixed TIT as pressure ratio is increased, exhaust temperature is reduced. This reduction in wasted heat raises the thermal efficiency of the cycle. An optimum pressure ratio for the simple cycle is reached when, for a given TIT, the benefits of the reduced exhaust temperature are counteracted by the increased compressor power needed to obtain the increased pressure ratio. This is shown in figure 2.1 where T' is the ratio of TIT to compressor inlet temperature and **r** is defined as the compressor pressure ratio. Gains in performance for the simple cycle have focused on increasing TIT and pressure ratio through the use of advanced materials, turbine blade cooling, and optimized

compressor design. Part load performance of the simple cycle is poor due to the dependence on operating at a high design point TIT.

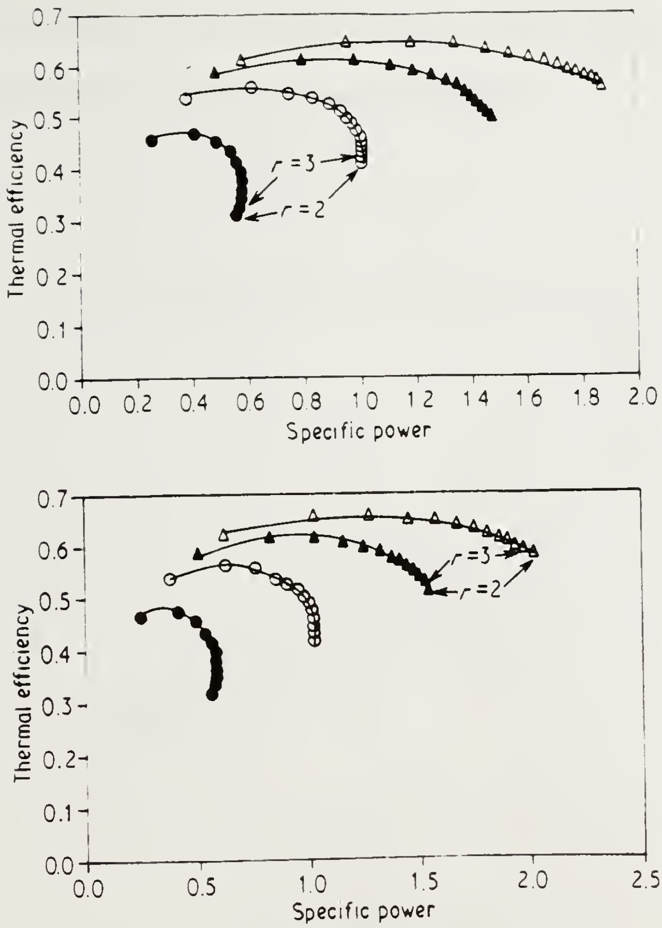
2.2 The Recuperated Cycle

The recuperated or heat-exchanger cycle is a modification of the CBE cycle which seeks to utilize waste heat in order to increase thermal efficiency. A recuperated-cycle gas turbine consists of a compressor, combustor, expander and heat exchanger and is designated CBEX. It is depicted schematically in figure 2.3. Since the heat exchanger extracts usable heat from the exhaust, thermal efficiency is increased at lower pressure ratios as shown in figure 3.1. Recuperation will be further discussed in chapter 3.



Plot of temperature against entropy and block diagram for heat-exchanger cycle

Figure 2.3 CBEX Schematic and T-s Diagram [18]



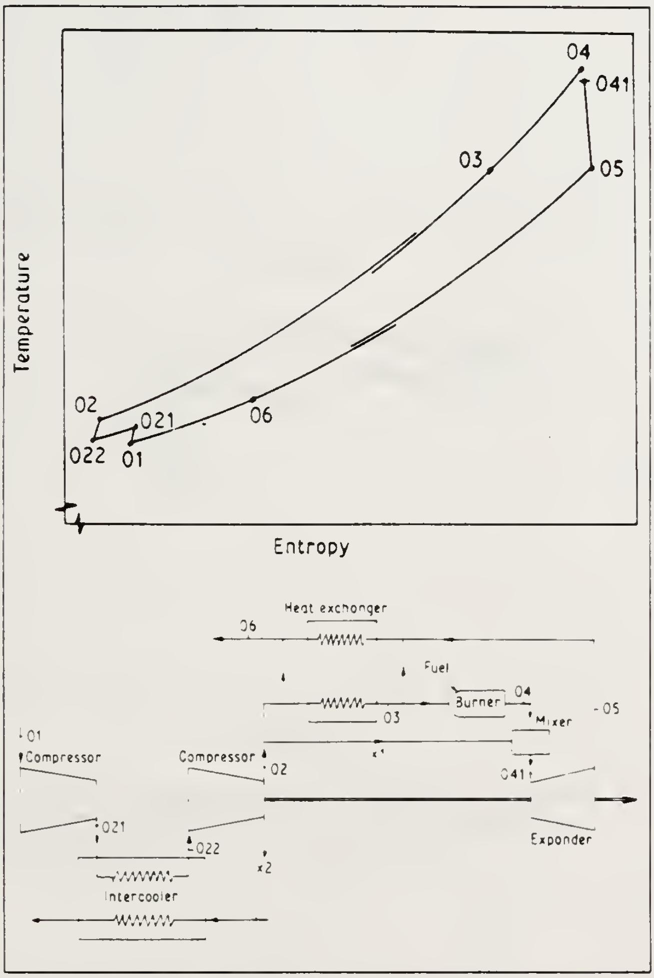
Plot of heat-exchanger cycle thermal efficiency agains specific power for large (upper) and small (lower) cooling-mass flow fraction. ●, $T' = 4$; ○, $T' = 5$; ▲, $T' = 6$; △, $T' = 7$

Figure 2.4 Thermal Efficiency vs. Specific Power - CBEX [18]

2.3 The Intercooled Recuperated Cycle

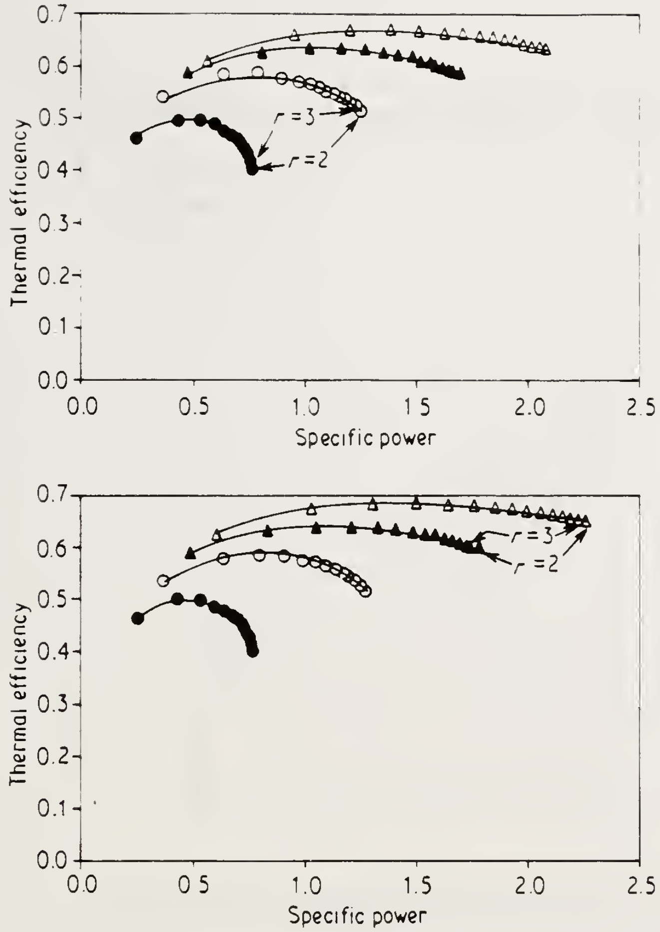
As discussed previously, as pressure ratio is increased so does the work necessary to drive the compressor. Power required to compress a working fluid is proportional to the initial temperature; thus, if the working fluid can be cooled between stages of compression, the overall power required for compression is reduced. The device which performs this, called an intercooler, coupled with a heat exchanger to take advantage of turbine exhaust waste heat, increases the overall thermal efficiency and specific power of the engine. This

recuperated cycle or CICBEX cycle is shown in figure 2.5 and performance is depicted in figure 2.6. Intercooling will be further discussed in chapter 3.



Plot of temperature against entropy and block diagram for intercooled cycle

Figure 2.5 CICBEX Schematic and T-s Diagram [18]



Plot of intercooled cycle thermal efficiency against specific power for large (upper) and small (lower) cooling-mass-flow fraction. ●, T' = 4; ○, T' = 5; ▲, T' = 6; △, T' = 7

Figure 2.6 Thermal Efficiency vs. Specific Power - CICBEX [18]

2.4 The Intercooled Exhaust-Heated Cycle

The intercooled exhaust-heated cycle is a slight variant of the intercooled recuperated cycle in that the combustor has now been placed after the expander. The cycle is shown in figure 2.6 with a T-s diagram and is designated CICXEB. For this study the rotary regenerator (figure 2.7) will perform as the heat exchanger in the cycle.

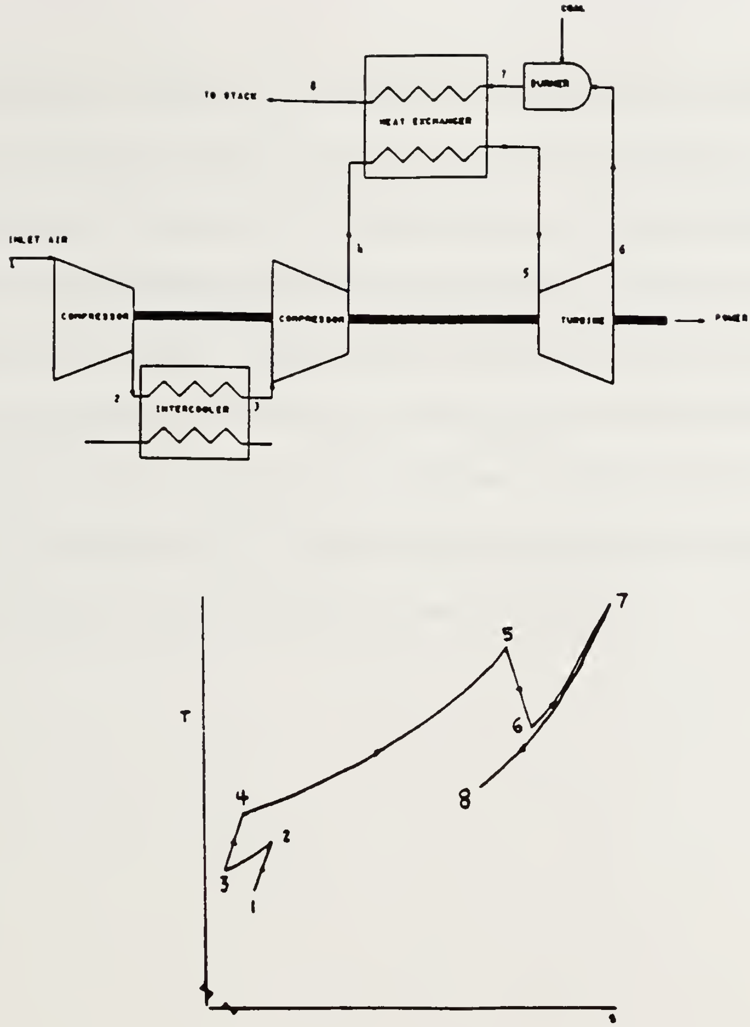


Figure 2.7 CICEBX Schematic and T-s Diagram [16]

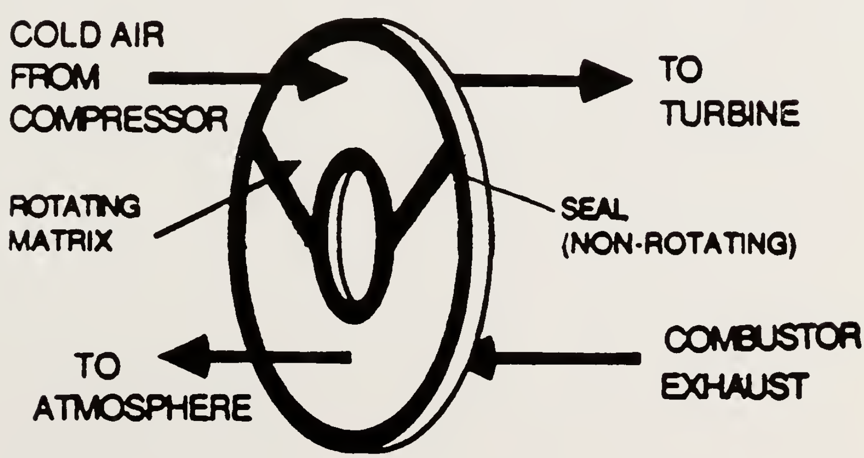


Figure 2.8 Rotary Regenerator [19]

The specific-power curves for the CICBEX and CICXEB are similar with the only difference in performance due to the transposition of the combustor which must now transfer heat energy to the cycle via a heat-exchanger. In fact, performances would be identical if the effectiveness of the heat-exchanger were 100% and all other component efficiencies were the same. The advantage of this cycle is that the expander never encounters the products of combustion so that fuel quality never becomes an issue for the turbine. The choice of the ceramic rotary regenerator and low cycle pressure ratio ensures maximized heat-exchanger effectiveness and minimized mass flow losses. Optimizing the pressure ratio for this cycle will also be discussed in chapter 5.

3.0 APPLICABLE TECHNOLOGIES AND DESIGN PHILOSOPHY

This chapter discusses in depth the advantages of recuperation, intercooling and utilization of variable-area power-turbine nozzles in maximizing both design and part-load performance. The integration of these technologies and maximized performance at relatively low pressure ratios is the design philosophy of the commercial engine conversion into the CICXEB cycle using coal as a primary fuel.

3.1 High-Efficiency Complex Systems

As the industrial use of the gas turbine has expanded, specific engine designs have evolved which are not aero-derivative in nature. These designs have prioritized thermal efficiency, part-load performance and specific fuel consumption. For industrial applications, the compactness of the engine has been sacrificed in order to gain these objectives. The general sizes for heat exchangers used for intercooling and heat recuperation are often many times larger than the engine itself. These new-generation engines of higher efficiency have several common characteristics. They employ low pressure ratios, high-effectiveness heat exchangers for heat "regeneration" and, possibly, intercooling. These cycle alterations are compared in figure 3.1.

Recuperation of the heat in the exhaust gases of the gas turbine provides for a reduction in the combustor temperature rise and thus a reduction in the amount of heat added to the engine or reduced fuel requirements. Intercooling is the process of removing heat between the stages of a multiple-stage compressor and results in less power utilized for operation of the compressor.

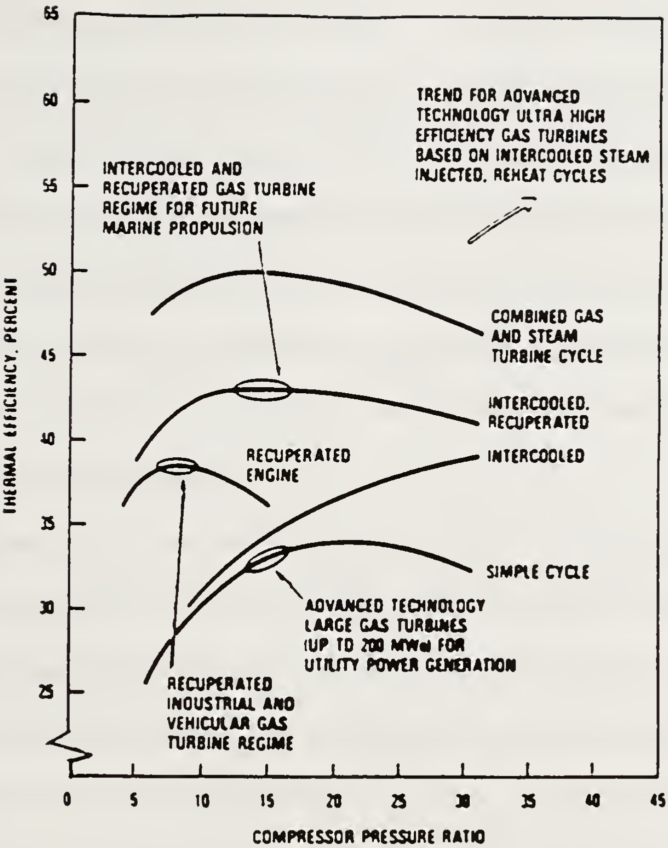


Figure 3.1 Effect of Technologies on Modern Gas Turbine Plants [20]

3.2 Recuperation

Recuperation is illustrated in figure 3.2. The shaded portion QR represents exhaust heat that is normally rejected from the engine. The temperature at the end of the compression process (point 2) is the limiting temperature at which heat is transferred into the the cycle. The effectiveness of a recuperator is a measure of how efficient the heat exchanger is at transferring this exhaust heat to the heat-addition portion of the cycle and thus replacing fuel as a source of heat. From this diagram, effectiveness is defined mathematically as:

$$\varepsilon_X \equiv \frac{T_a - T_2}{T_4 - T_2}$$

Therefore, if a heat exchanger could provide perfect recuperation, the temperature of the working fluid entering the combustor at point "a" would be the same as the exhaust gas temperature leaving the power turbine at point 4. Flow restrictions in the recuperator cause pressure losses on both the "hot" and "cold" sides of the heat exchanger and result in a loss in net work at all pressure ratios as compared with the simple cycle. Since the compression process also increases the temperature of air, recuperation is possible only at compression pressure ratios below the level at which the temperatures at the end of compression (point 2) and expansion (point 4) are equal. This optimum pressure ratio increases as the firing temperature or TIT is increased [21].

Heat exchangers which are stationary are referred to as recuperators while those in which the flow is periodic are called regenerators. Generally, higher effectiveness is achieved by designing heat exchangers with greater heat-transfer areas. For the overall size of the component to remain small, the hydraulic diameter of the passages within the heat exchanger should also remain small [5]. Generally, increasing heat-transfer area also increases the pressure losses within the heat exchanger due to greater flow restrictions. Both the pressure drop and effectiveness must be considered in determining which type of heat exchanger will result in maximum thermal efficiency for the cycle [23]. The cost of the of the heat exchanger must also be weighed against the projected fuel savings during the life cycle of the project. Adding heat regeneration to a cycle will increase thermal efficiency but decrease the net work as compared to the baseline simple cycle. A regenerative cycle also has the added advantage of better part-load performance due to increased heat-exchanger effectiveness at part load operation. Figure 3.3 illustrates the variation of thermal efficiency with power output for a hypothetical cycle with and without a heat exchanger.This figure assumes a lossless heat exchanger but serves to illustrate the effects of recuperation on a cycle.

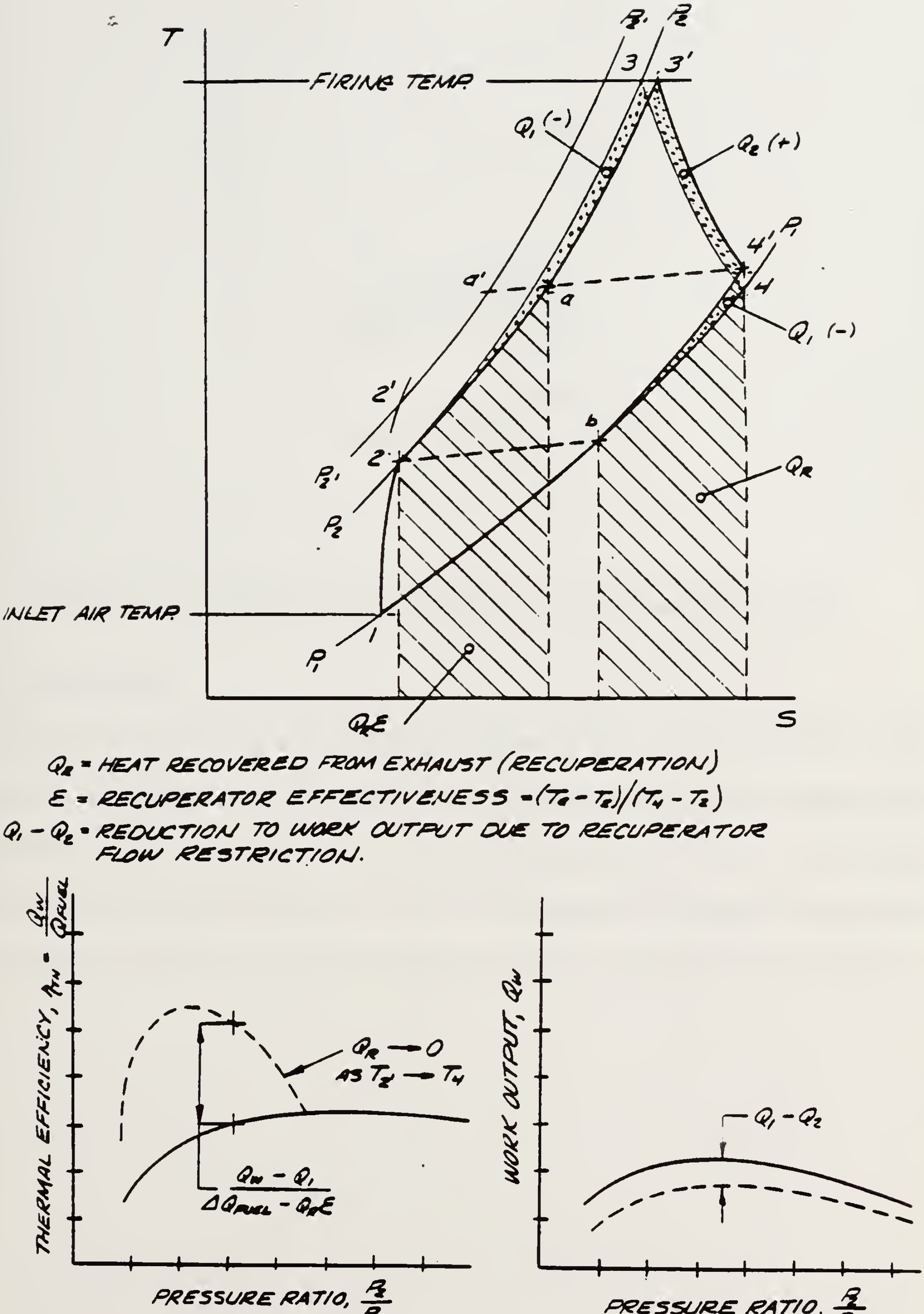


Figure 3.2 Brayton Cycle - Effect of Recuperation [21]

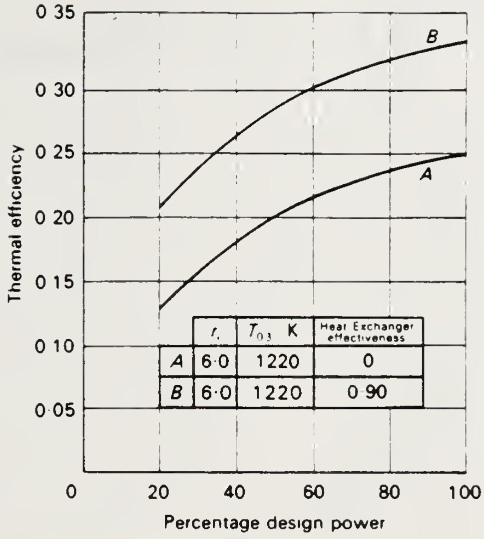


Figure 3.3 Recuperated and Simple Cycle Comparison [22]

3.3 Intercooling

The process of intercooling is illustrated in figure 3.4 and is applicable only in cycles that employ multi-stage compression and is performed between the stages of compression. For simplicity it is illustrated for a two stage compression process. Heat is rejected to the intercooler along the process denoted as "a-b". This heat rejection reduces the temperature and increases density of the working fluid prior to it entering the next stage of

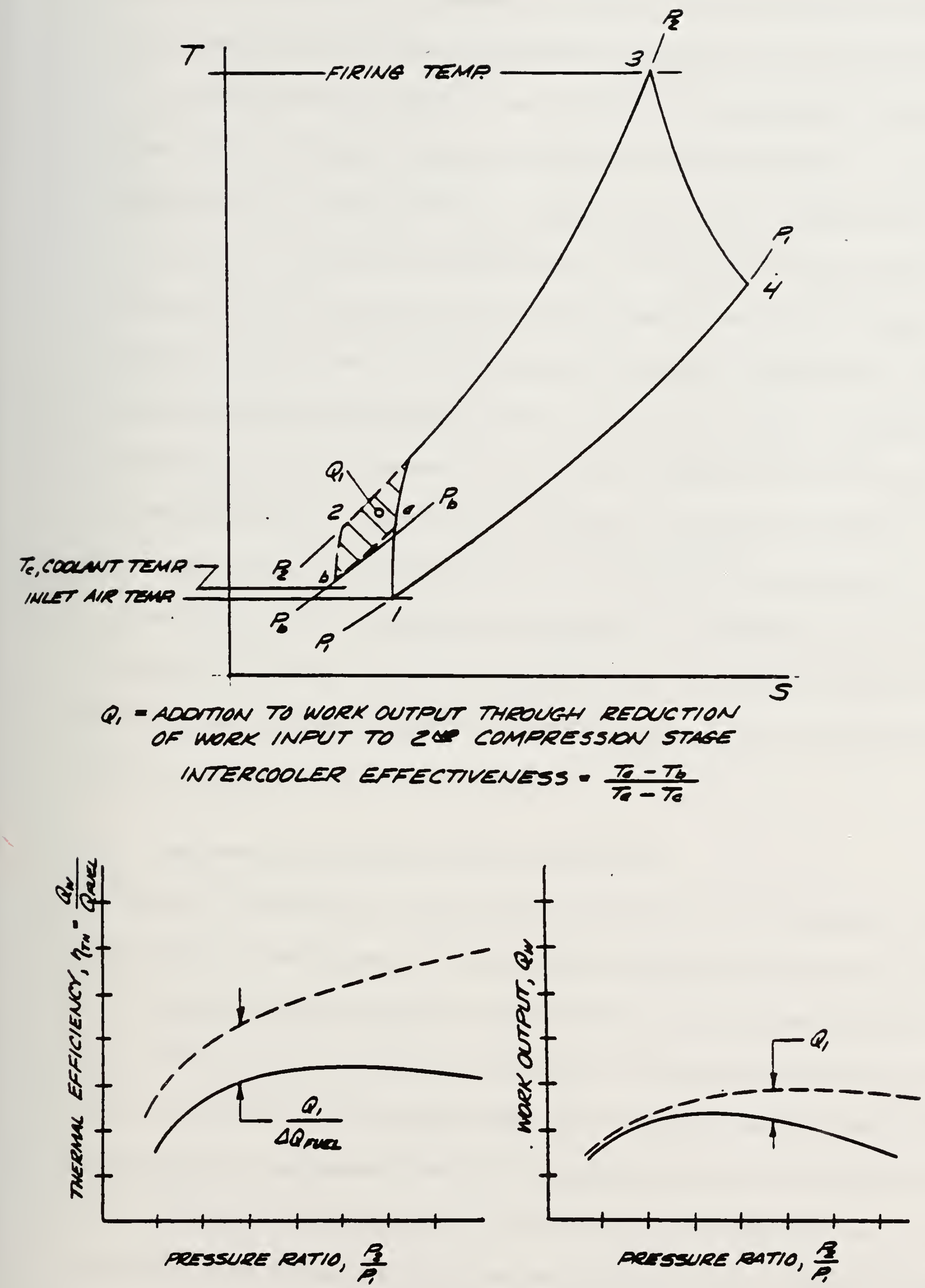


Figure 3.4 Brayton Cycle - Effect of Intercooling [21]

compression and results in a reduction of work input to the second stage of compression which is represented as a positive Q1 and added to the work output of the cycle [21].

In a gas turbine the compressor may be driven by a compressor (or gas-producer) turbine as in a split-shaft arrangement similar to the GE LM2500. In a single shaft configuration, like the Allison 501-K, a single turbine provides compressor power as well as shaft power. In either case compressor work is not useful work and reduces the amount of energy that may be extracted from the cycle for shaft power. Pressure losses in ducting between the stages that lead to the intercooler are also a consideration for axial-flow compressors whereas intercooler-ducting losses in a centrifugal compressor may be minimized due to the radial-inward-outward flow between the stages which is more accommodating to this modification. Intercooling is particularly attractive for marine applications as there is an unlimited source of cooling fluid for the intercooler.

Intercooling results in a reduction in compressor work and, therefore, an increase in net work output as well as an increase in thermal efficiency. The increase in efficiency is small at lower pressure ratios and grows continuously as pressure ratio is increased because more heat is available for removal by the intercooler.

3.4 Combining Intercooling and Recuperation

Figure 3.5 illustrates the combined effects of recuperation and intercooling. These two technologies are complementary in that the reduced temperature at the end of compression (T'2 versus T2) provides a larger temperature differential for heat transfer from the exhaust gases. ΔQ_R represents this substantial increase to the heat transferrable from the exhaust over the non-intercooled recuperated cycle. This results in an increased thermal efficiency in addition to the increase provided by the reduction in necessary compressor work due to intercooling alone. The optimum pressure ratio at maximum thermal efficiency is increased in addition to the increase provided by the reduction in necessary compressor work due to intercooling alone. The optimum pressure ratio at maximum thermal efficiency is increased

over the non-intercooled recuperated cycle and, in fact, the recuperation of gas-turbine engines of high pressure ratio is feasible only in conjunction with intercooling because of the low turbine-outlet temperatures.

3.5 Optimum Pressure Ratios

When looking at the preliminary design of a recuperated or intercooled- recuperated gas-turbine engine the priorities placed on maximum thermal efficiency or specific work may drive the final design point of the cycle. For maximum efficiency the recuperated engine would be designed at a slightly lower pressure ratio than a similar intercooled-recuperated cycle. In a previous study by Wilson [17] which employed a ceramic rotary regenerator, the optimum pressure ratio for the regenerative cycle was found to be approximately 3:1 and that for the intercooled-regenerative engine was approximately 4:1. These values were determined for maximum efficiency. With any design the particular characteristics of the heat exchangers may determine a limiting pressure ratio based on component losses.

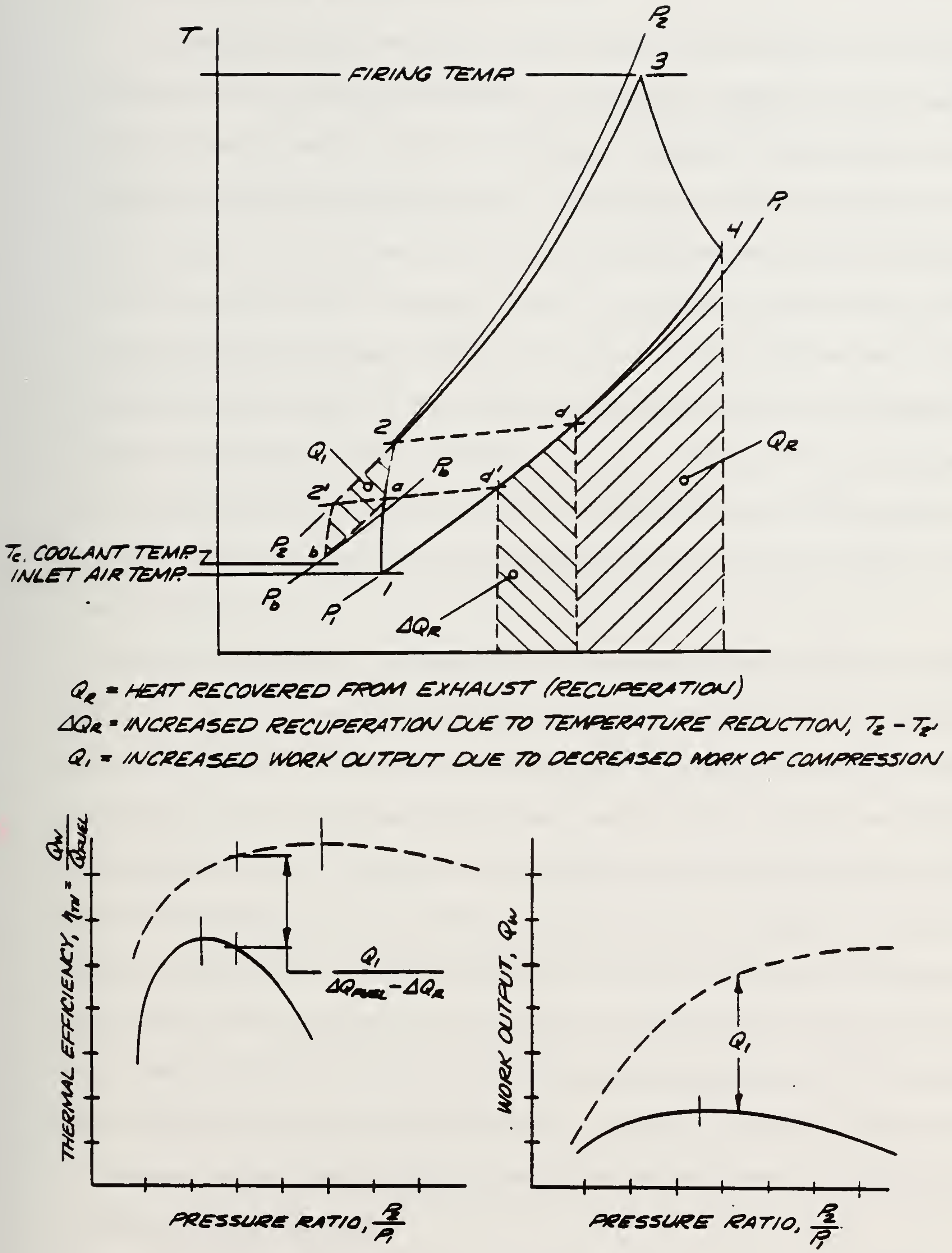


Figure 3.5 Recuperated Brayton Cycle - Effect of Intercooling [21]

3.6 Variable-Area Power-Turbine Nozzles

As stated previously, one of the disadvantages of the simple cycle is part-load performance. Intercooling and recuperation, in combination, greatly improve part and full-load performance.Another current technology, variable-area power-turbine nozzles, specifically enhance part-load performance. For the simple cycle, thermal efficiency was dependent upon TIT and the basic cause for poor part-load performance is the rapid drop of TIT with decreasing power. As shown in figure 3.3, recuperation greatly enhances the part and full-load thermal efficiency of the simple cycle, but the curve is shifted only vertically and the basic shape remains the same. This diagram assumes a constant effectiveness for a lossless heat exchanger at all loads but in fact part-load performance of a heat exchanger is often slightly better than at design point. Maintaining a maximum TIT throughout the operating range of the engine is, then, the desired goal and the major objective of the variable nozzles.

Variation of throat area is performed by turbine nozzles that rotate about an axis (figure 3.6) [24]. When the load on a split-shaft gas turbine is reduced below its maximum-efficiency rating, the mass flow of the working fluid is reduced because of a reduction in compressor speed. In a standard fixed-geometry engine the TIT is reduced and, therefore, thermal efficiency. If flow capacity can be altered for the power turbine at various loadings then TIT and efficiency can be maintained as high as possible during part-load operation. Variable turbine nozzles would decrease the the throat area of the power turbine resulting in an increase of the overall fraction of the combined expansion pressure ratio. This would then control the pressure ratio across the compressor turbine and allow for the higher desired TIT for the power turbine because of a smaller temperature drop across the compressor turbine. The combined expansion ratio of the turbines is explained in the equation on the following page.

$R_{exp}=r_{ct}\, r_{pt} = R_c(1-\Delta p/p)$

r_{ct} :pressure ratio of compressor turbine

r_{pt}: pressure ratio of power turbine

R_{exp}: pressure ratio of expansion

R_{comp}: pressure ratio of compressor

$\Delta p/p$: total engine pressure losses

If the power turbine pressure ratio is increased then the compressor turbine pressure ratio decreases resulting in a TIT which is maintained at a maximum over all operating conditions. This effect is shown in figure 3.7 and depicts the extent to which compressor speed may be reduced without degrading TIT. This is very much dependent on the compressor surge characteristics [25]. Another benefit of operating near the compressor surge line is the likelihood of an increase of compressor efficiency. Both of these effects will improve part load performance. Variable nozzles also tend to degrade power turbine efficiency at design speeds but it has been shown that this loss in component efficiency can be more than offset by maintaining a higher TIT at part load with area variations ranging from +20% to -20% [26].

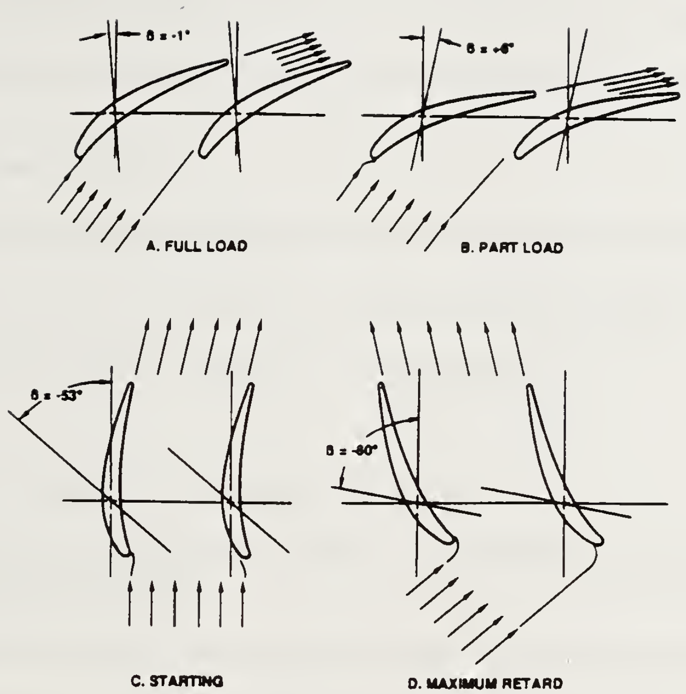


Figure 3.6 Nozzle Vane Positions [24]

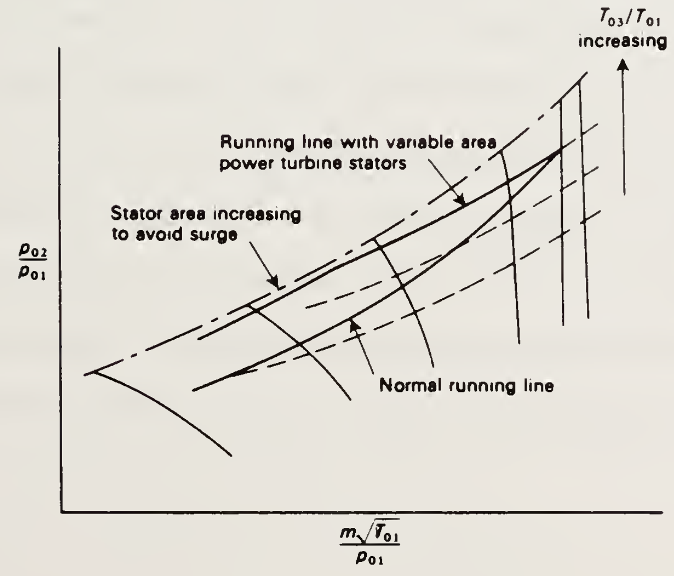


Figure 3.7 Effect of Variable Power Turbine Stators on Running Line [20]

4.0 ENGINE SELECTION

The engine chosen for conversion is the Solar 5650 industrial gas turbine. This section documents the engine-selection process as well as the engine features and performance.

4.1 The Selection Process

Several criteria were established to narrow down the number of candidate engines to a manageable size. The criteria are based on requirements used to size the "blue-sky" engine [16] and minimize the conversion expense . The criteria are:

1. the power output must be approximately 2 MW,
2. the turbine-inlet temperature should be about 1300 K for high thermal efficiency with low ash stickiness,
3. performance and relevant design information must be readily available,
4. a low-pressure-ratio cycle is preferred,
5. a two-stage centrifugal compressor would facilitate intercooling, and
6. the engine must be developed or in production.

The matrix of candidate engines and their basic features shown in Table 4.1 was created from [27] and [28]. Numerous other engines were eliminated from consideration for various reasons. The Solar 5650 emerged as the clear choice for conversion to an exhaust-heated, coal-burning engine since few production engines are specifically designed for industrial, low-pressure-ratio operation. Although Solar was reluctant to provide design and performance details, several non-proprietary reports were discovered to contain all the relevant information needed to carry out this preliminary design study.

Table 4.1 Candidate Gas Turbines [27, 28]

Manufacturer Model	Output (kW)	PR	Flow (kg/s)	T.I.T. (K)	ηth (%)	Speed (RPM)
AVCO-Lycoming						
TF25	1865	6.9	9.6	---	23	14500
Ruston						
TA2500	1865	5.1	12.9	1124	21.2	7950
IHI						
IM 100-4G	1110	8.4	6.4	1018	23	19500
Kawasaki						
M1A-03	1470	9.2	9.1	---	20.1	22000
Dresser-Rand						
KG2	1550	4	13.1	1100	16.7	18100
Solar Gas Turbine						
Centaur	2945	9.0	17.3	1150	25	15700
5650	2768	6.5	17.2	1241	33.5	10620
Yanmar						
AT270C	2400	8.1	15.4	1173	---	1800
Pratt & Whitney						
SPW 124	1790	13.7	7.7	---	---	20000
General Electric						
LM500	3730	---	15	---	---	7000

4.2 The Solar 5650 Features

The Solar 5650 industrial gas turbine has been in development for twenty years. Solar and its parent company, Caterpillar Tractor, designed the 5650 to compete with large diesel engines. It was a proposed replacement for the less-efficient Allison 501-K currently used aboard U. S. Navy ships as a generator set. Although the 5650 is not in full-scale production, several pilot sites currently use the 5650 for full-or part-time power generation (see figure 4.1) [29].

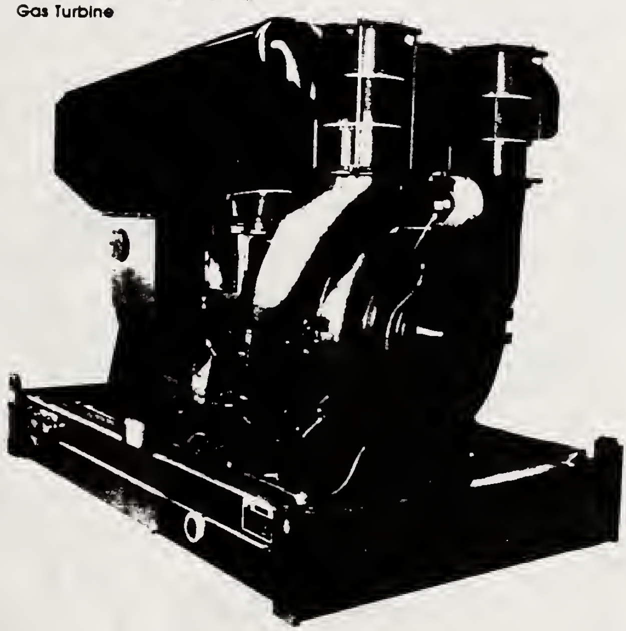


Figure 4.1 Solar 5650 Industrial Gas Turbine [18]

The 5650 is a twin-shaft, low-pressure-ratio, recuperated gas turbine with several unique features. The overall dimensions of the engine are listed in Table 4.2. The modular engine components consist of a primary-surface recuperator, two-stage centrifugal compressor, annular combustor, single-stage, air-cooled gas-producer turbine and a single-stage power turbine with variable inlet vanes (see figure 4.2). The modular primary-surface folded-sheet-metal recuperator is the most innovative feature. The elements of the high-effectiveness recuperator can slide relative to each other thus avoiding thermal stress and strain. Unfortunately, this engine component cannot economically be used in the exhaust-heated design due to inaccessibility for cleaning after fouling,. The variable-area power-turbine nozzle allows quick load response and high part-power thermal efficiency for reasons discussed in section 3.6. The turbine-inlet temperature is hot enough to attain high thermal efficiency yet low enough to reduce ash stickiness in the regenerator. The manufacturer claims that with improved turbine-blade cooling the 5650 is capable of a 98 degrees K increase in turbine-inlet temperature while still meeting the 100,000-hour design life [29].

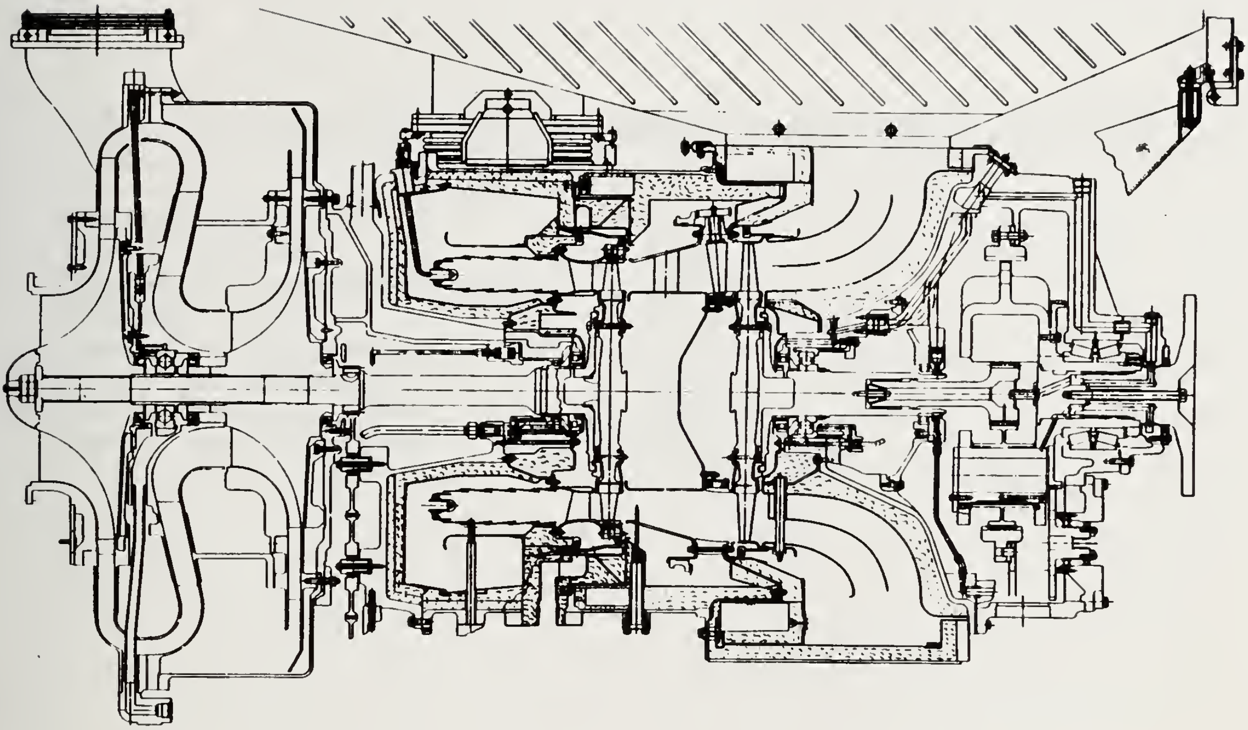


Figure 4.2 Solar 5650 Engine Cross-Section [30]

Table 4.2 Engine Dimensions [30]

Physical Dimensions	Engine	Engine Including Base
Length (m)	2.896	3.659
Width (m)	1.930	2.090
Height (m)	2.235	2.730
Weight (kg)	7264	10215

4.3 Solar 5650 Performance

Several sources of quoted design-point performance of the Solar 5650 appear to be in conflict [21,24,29,30]. The minor inconsistencies are attributed to the fact that the 5650 is currently not in production and is constantly undergoing new performance evaluations. The engine performance in Table 4.3 is believed to be from the most recent testing.

Table 4.3 Base Solar 5650 Design-Point Performance (sea-level, 288 K) [21,24,29,30]

Thermal Efficiency (%)	33.5
Power Output (kW)	2768
Specific Fuel Consumption (kW/kg-hr)	0.2506
Mass Flow (kg/s)	17.22
Compressor Pressure Ratio	6.37
Compressor Speed (RPM)	13100
Turbine-Inlet Temperature (K)	1241.3
Power-Turbine Speed (RPM)	10620

5.0 ENGINE CONVERSION

The engine cycle analysis and three design options to achieve the optimal-cycle performance with the intercooled base Solar 5650 are presented in this section.

5.1 Analysis Overview

The analytical procedure to modify the intercooled base Solar 5650 to an intercooled exhaust-heated, coal-burning engine consists of two primary tasks: cycle analysis and turbomachinery preliminary design. The cycle analysis determines the overall performance of the modified engine and sets the thermodynamic requirements to which the turbomachinery must be designed. A brief summary of the steps followed to carry out these tasks is included here. A detailed explanation of each step is given in later sub-sections.

1. The predicted design-point performance of the intercooled 5650 was matched with the performance calculated by cycle-analysis computer programs.

2. The design-point cycle-analysis computer program was modified to simulate the intercooled exhaust-heated, coal-burning Solar 5650 engine.

3. The optimal compressor pressure ratio for the intercooled, exhaust-heated, coal-burning Solar 5650 was determined.

4. Three alternatives to achieve optimal performance for the intercooled, exhaust-heated 5650 were examined. The options are:

   a. run the modified engine at its baseline pressure ratio,
   b. design all-new turbomachinery and run at an optimized pressure ratio, or
   c. make no turbomachinery modifications, just decrease engine speed.

The merits of each of the various options listed above are judged on an overall life-cycle-cost basis. It is important to note that Solar's intercooled version of the 5650 engine did not alter the turbomachinery in any way except for the addition of intercooling between the stages of the two-stage centrifugal compressor. The capital cost of the converted engine is minimized by purposely holding the turbine-inlet temperature and mass flow close to the

intercooled base 5650 levels. This ensures that the redesigned turbomachinery will remain at approximately the same size and shape as the base 5650 turbomachinery. The power output, engine life, engine bearings and accessories will also remain nearly constant. Performance and cost of components used to provide intercooling were taken from Karstensen [20].

The preliminary design of an adequate combustor and engine control system is considered beyond the scope of this report. A promising coal-burning-combustor technology is in development (slagging, two-stage combustors) and is assumed acceptable for the dry-micronized coal to be burned in this engine and is briefly discussed in a later section. Similarly, aeromechanical and stress analysis of the redesigned turbomachinery is regarded as too detailed for the purposes of this feasibility study.

5.2 Base Solar 5650 Performance Data-Match

The data-match of the predicted intercooled 5650 design-point performance served as the stepping stone from which the performance of the exhaust-heated, coal-burning engine was extrapolated. The CYCLE computer program written by Tampe [16] for the "blue-sky" design was modified to represent the recuperated cycle of the intercooled Solar 5650 . The intercooled 5650 mass flow, compressor-pressure ratio, component efficiencies, cooling flows and duct losses were entered into the computer model. Mechanical losses and fuel properties were adjusted to match the measured performance. Polytropic component efficiencies, pressure and mass losses, remained the same for the model. Power turbine shaft losses were assumed to be 1.5% to provide an accurate match. The combustor efficiency was adjusted to 95% and appears unreasonably low but this value was also reported by Solar for the base 5650 engine [30]. Ducting and interstage pressure losses occurring in the compressor due to intercooling were assumed to be the same value as predicted by Karstensen [21]. Table 3.1 compares the predicted base intercooled 5650 cycle parameters to the parameters used in the data match. Due to the simplicity of the

computer model, the effectiveness of the recuperator was lowered slightly from the reported data in order to provide a consistent match. The compressor and turbine efficiencies listed in the table are polytropic.

Table 5.1 Intercooled Base 5650 Design-Point Cycle Parameters

	IC Solar Data [21]	IC Base 5650 Model
Compressor 1st Stage		
Efficiency (%)	84.1	84.1
Pressure Ratio	2.81	2.78
Mass Flow (kg/s)	18.77	18.77
Intercooler		
Effectiveness	.902	.902
ΔP (%)	.4	.4
Interstage ΔP (%)	3.61	3.61
Compressor 2nd Stage		
Efficiency (%)	79.6	79.6
Pressure Ratio	2.59	2.56
Mass Flow (kg/s)	18.77	18.77
Recuperator		
Effectiveness	0.887	0.865
Cold-Side ΔP (%)	3.53	3.53
Hot-Side ΔP (%)	6.29	6.29
Combustor		
ΔP (%)	4.3	4.3
Efficiency (%)	100.7	95 [30]
Gas-Producer Turbine		
Cooling Flow (% WA1)	2.5	2.5
Efficiency (%)	87.9	87.9
Duct Pressure Loss (%)	1.97	1.97
Power Turbine		
Cooling Flow (% WA1)	0.8	0.8
Efficiency (%)	86.9	86.9
Duct Pressure Loss (%)	6.0	6.0
Power Output Module		
Shaft Losses (%)	1-3	1.5
Fuel-Heating Value (kJ/kg)	unknown	42700

The comparison of overall performance is shown in Table 5.2, Excellent agreement was reached between the data predicted by Solar and model-calculated engine performance.

Table 5.2 Overall Performance
Intercooled Model vs. Predicted Intercooled Data

	IC Solar Data [20]	IC Base 5650 Model
Thermal Efficiency (%)	36.3	36.6
Power Output (kW)	3520	3520
Specific Power	.647	.647
Specific Fuel Consumption (kg/kW-hr)	.2317	.2312

Table 5.3 show a breakdown of the measured and calculated component performances. There are some subtle differences in component temperatures but this reflects the limitations of the computer model. In general, the individual component performance for the intercooled base engine is reasonably matched.

Table 5.3 Component Performance
Intercooled Model vs. Predicted Intercooled Data

	IC Solar Data [20]		IC Base 5650 Model	
	Inlet	Exit	Inlet	Exit
Compressor				
Temp. (K)	288.0	439.3	288.0	436.9
Flow (kg/s)	18.77	18.77	18.77	18.77
Pressure (kPa)	101.3	705.1	101.3	719.2
Recuperator Cold Side				
Temp. (K)	439.3	799.3	436.9	799.3
Combustor				
Temp. (K)	799.3	1244.7	799.3	1241.3
Flow (kg/s)	18.07	18.30	18.08	18.38
Gas-Producer Turbine				
Temp. (K)	1241.3	1025.9	1241.3	1023.8
Flow (kg/s)	18.39	18.39	18.38	18.85
Power Turbine				
Temp. (K)	1012.6	848.2	1013.6	855.9
Flow (kg/s)	18.84	18.84	18.85	19.00
Recuperator Hot Side				
Temp. (K)	844.8	512.6	855.9	502.0

An accurate design-point computer representation of the intercooled base Solar 5650 has been created to facilitate the performance prediction of the converted engine.

5.3 Exhaust-Heated Solar 5650 Cycle Performance Prediction

The intercooled base 5650 computer model provided a known foundation to which alterations could now be made to produce a model of the intercooled, exhaust-heated, coal-burning engine. The combustor was extracted from its original position between the recuperator exit and gas-producer-turbine inlet and a new slagging combustor was placed after the power turbine. The primary-surface recuperator was removed from the cycle and

replaced with a ceramic rotary regenerator. Rotary-regenerator-sizing and performance logic was added to the program.

The regenerator-sizing procedure is outlined in Wilson [5] and detailed performance equations are derived in Hagler [19]. The algorithms were developed and programmed by Tampe [16] for the "blue-sky" design. The surface geometry of the ceramic-regenerator matrix chosen for this exhaust-heated application is summarized in Table 5.4. The programs developed for the intercooled, exhaust-heated, cycle are similar to those developed with Nahatis [31] for the non-intercooled, exhaust-heated cycle.

Table 5.4 Ceramic Matrix Surface Geometry [5]

Stanford Univ. Core Number	503A
Passage Count (No./in^2)	1008
Hydraulic Diameter (microns)	511
Area Density (m^2/m^3)	5551
Porosity	0.708
Solid Density (kg/m^3)	2259

The 503A matrix was selected based on sensitivity studies conducted by Tampe [16]. More recent information indicates that a matrix with a larger hydraulic diameter would decrease susceptibility to deposition and reduce the axial temperature gradient. A later section will compare the sizing changes necessary for cores with different hydraulic diameters. For all cases the effectiveness of the regenerator is selected to be 0.975. Figure 5.1 illustrates a scaled drawing of a probable engine cross-section which shows diameters and thicknesses for the 503A matrix. If a matrix with a larger hydraulic diameter had been chosen, the thickness would have been larger. The regenerator dimensions and design-point performance as calculated in the computer model are shown in Table 5.5. Two medium-sized regenerators are used rather than one large regenerator or many small regenerators.

Table 5.5 Regenerator Dimensions and Performance Intercooled Exhaust-Heated 5650 Model

Number of Disks	2
Core Type	503A
Effectiveness	0.975
Cycle Pressure Ratio	7.10
Diam. of Each Disk (m)	3.5198
Thickness of Each Disk (m)	0.1386
Mass of Each Disk (kg)	853.7
Rotational Speed (RPM)	1.74
Power Consumption (kW)	11.62
Total Radial Seal Leakage (% WA1)	3.22
Total Circumf. Seal Leakage (% WA1)	1.41
Cold Side	
Pressure drop (%)	.17
Heat-transfer area (m^2)	1467.9
Free-face area (m^2)	1.351
Face area (m^2)	1.908
Hot Side	
Pressure drop (%)	3.11
Heat-transfer area (m^2)	4280.3
Free-face area (m^2)	3.940
Face area (m^2)	5.565

The component losses and efficiencies in the exhaust-heated, coal-burning 5650 model were assumed the same as those presented in the base 5650 model with a few noted exceptions. The recuperator pressure losses were eliminated and replaced with the calculated regenerator pressure losses. The additional ducting traveling to and from the two regenerators was assumed to add a 2.0 % pressure loss to the cycle and the efficiency of the slagging combustor was set at 95% [2]. The fuel-heating value was lowered to 34262 kJ/kg to simulate the energy available in West Virginia, low-volatility-bituminous coal. The ultimate analysis of the coal in Table 5.6 shows that this coal has a relatively low (4%) ash content.

Table 5.6 Coal Ultimate Analysis [16]

Species	Moisture	C	H	S	O	N	Ash
% wt.	2.7	84.7	4.3	0.6	2.2	1.5	4.0

The overall predicted design-point performance of the intercooled, exhaust-heated, coal-burning 5650 engine is compared to the intercooled base 5650 in Table 5.7. The small performance penalty in converting from one configuration to the other is due to the regenerator leakages and the sub-optimal compressor pressure ratio. The relative life-cycle cost of this conversion will be examined later.

Table 5.7 Overall Performance Comparison
Intercooled Base Model vs. Intercooled Exhaust-Heated Model

	IC Base 5650 Model	IC Exhaust-Heated 5650 Model
Thermal Efficiency (%)	36.6	35.8
Power Output (kW)	3520	3257
Specific Power	.647	.599
Specific Fuel Consumption (kg/kW-hr)	.2312	.2938

The component performance of the exhaust-heated model versus the base model is shown in Table 5.8 . Note that "heat exchanger" in the tables denotes the recuperator for the intercooled base 5650 and the rotary regenerator for the intercooled, exhaust-heated 5650. The component performance reflects the physical changes made to the base 5650.

Table 5.8 Component Performance
Intercooled Base Model vs. Intercooled Exhaust-Heated Model

	IC Base 5650 Model		IC Exhaust-Heated 5650 Model	
			Inlet	Exit
Compressor				
Temp. (K)	288.0	436.9	288.0	436.9
Flow (kg/s)	18.77	18.77	18.77	18.77
Pressure (kPa)	101.3	719.2	101.3	719.2
Heat-exchanger Cold Side				
Temp. (K)	436.9	799.3	436.9	1244.3
Combustor				
Temp. (K)	799.3	1241.3	851.5	1265.0
Flow (kg/s)	18.08	18.38	17.93	18.18
Gas-Producer Turbine				
Temp. (K)	1241.3	1023.8	1244.3	1017.0
Flow (kg/s)	18.38	18.85	17.31	17.78
Pressure (kPa)	636.5	261.0	689.36	271.4
Power Turbine				
Temp. (K)	1013.6	855.9	1007.0	851.5
Flow (kg/s)	18.85	19.00	17.78	17.93
Pressure (kPa)	255.8	119.0	266.05	124.6
Heat-Exchanger Hot Side				
Temp. (K)	855.9	502.0	1265.0	483.4

The proposed conversion from the intercooled base 5650 to the exhaust-heated, coal-burning 5650 engine modeled above and shown in figure 5.1 would consist of five basic steps:

1. remove the existing annular combustor and insert smooth ducting in its place;
2. remove the primary-surface recuperator;
3. insert two ceramic rotary regenerators in parallel between the compressor and gas-producer turbine;

4. connect the inlet of a slagging combustor to the power-turbine exhaust and the outlet of the combustor to the rotary regenerator hot-side inlets; and
5. duct the compressor-exit air into the regenerator cold-side.

The overall performance of the intercooled, exhaust-heated 5650 shown in Table 5.7 may be improved by optimizing the compressor-pressure ratio. The data in the table are calculated at the intercooled base 5650 design point. A new, optimized design point for the exhaust-heated 5650 model must now be derived.

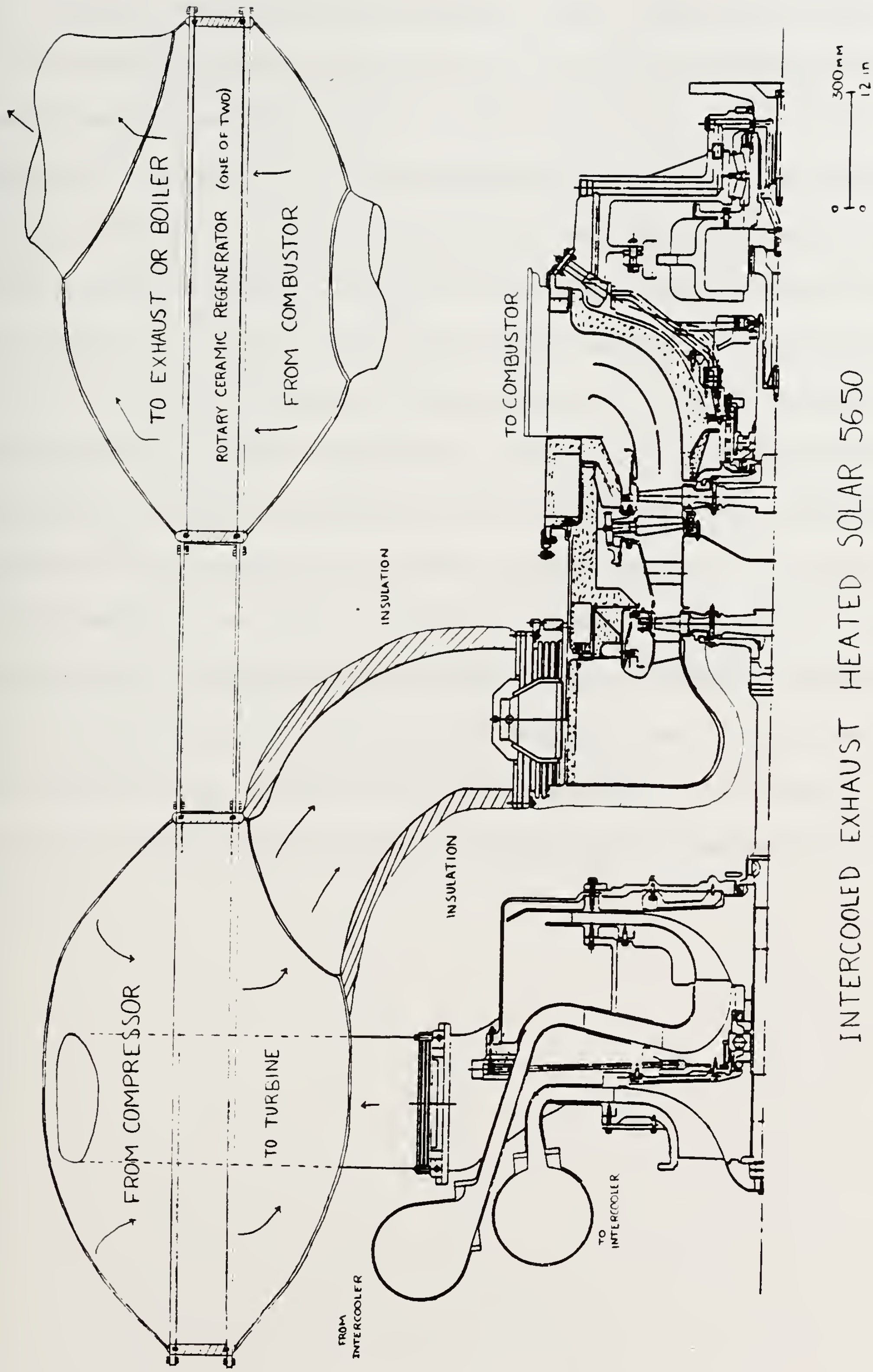


Figure 5.1 Intercooled Exhaust-Heated 5650

5.4 Optimal Intercooled Exhaust-Heated 5650 Cycle Pressure Ratio

The optimal cycle pressure ratio occurs between the points where thermal efficiency and specific power are near their peak values. These parameters cannot be maximized simultaneously, but they can be optimized approximately by constructing a curve of thermal efficiency versus specific power and choosing the pressure ratio at which the percentage decrease in thermal efficiency is greater than the percentage increase in specific power[2] . Each point on this curve represents the design point of a different engine, but each engine has the same turbine-inlet temperature and mass flow rate. The engine component efficiencies change slightly with pressure ratio according to the relationships defined in Wilson [5]. The plot of thermal efficiency versus specific power were constructed using the intercooled exhaust-heated 5650 computer model (see figure 5.2). The optimal pressure ratio for the intercooled exhaust-heated cycle was chosen as 4.5 while the optimal pressure ratio for the non-intercooled exhaust-heated cycle was chosen in a previous study done by Nahatis [31] as 4.0. The overall performance of the optimal cycle is compared to the original intercooled, exhaust-heated 5650 cycle in Table 5.9. The component performance of the optimal cycle is shown in Table 5.10. Regenerator size is listed in Table 5.11.

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[2] A rigorous optimization would require calculation of the life-cycle costs over a range of pressure ratios.

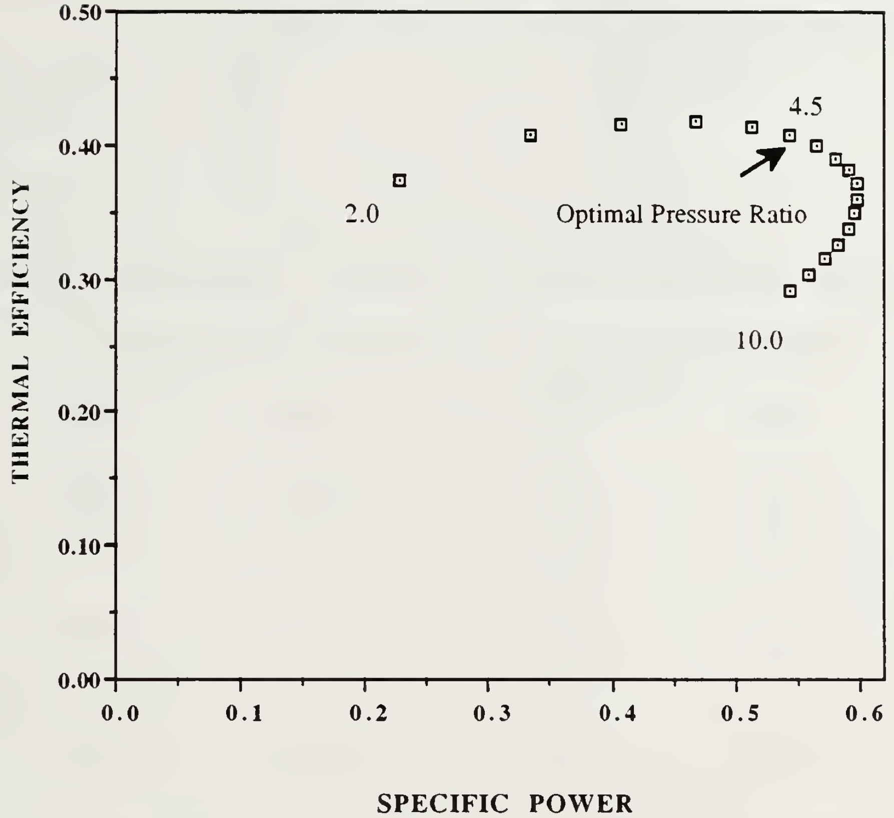


Figure 5.2 Design-Point Thermal Efficiency vs. Specific Power (Intercooled Exhaust-Heated Cycle)

Table 5.9 Overall Performance Comparison
Intercooled Exhaust-Heated Model vs. Optimal Intercooled Model

	IC Exhaust-Heated 5650 Model	Optimal IC Exhaust-Heated 5650 Model
Thermal Efficiency (%)	35.8	40.8
Power Output (kW)	3257	2961
Specific Power	.599	.545
Specific Fuel Consumption (kg/kW-hr)	.2938	.2577

Table 5.10 Component Performance Comparison
Intercooled Exhaust-Heated Model vs. Optimal Intercooled Model

	IC Exhaust-Heated 5650 Model		IC Optimal Exhaust-Heated 5650 Model	
	Inlet	Exit	Inlet	Exit
Compressor				
Temp. (K)	288.0	436.9	288.0	373.7
Flow (kg/s)	18.77	18.77	18.77	18.77
Pressure (kPa)	101.3	719.2	101.3	455.9
Intercooler				
Temp. (K)	406.7	312.6	373.7	309.4
Heat-exchanger Cold Side				
Temp. (K)	436.9	1244.3	395.6	1243.3
Combustor				
Temp. (K)	851.5	1265.0	940.10	1265.0
Flow (kg/s)	17.93	18.18	18.24	18.44
Gas-Producer Turbine				
Temp. (K)	1244.3	1017.0	1243.3	1086.1
Flow (kg/s)	17.31	17.78	17.62	18.09
Pressure (kPa)	689.36	271.4	436.91	234.52
Power Turbine				
Temp. (K)	1007.0	851.5	1075.4	940.1
Flow (kg/s)	17.78	17.93	18.09	18.24
Pressure (kPa)	266.05	124.6	229.90	124.88
Heat-Exchanger Hot Side				
Temp. (K)	1265.0	483.4	1265.0	449.6

Table 5.11 Comparison of Regenerator Dimensions and Performance Intercooled Exhaust-Heated Model vs. Optimal Intercooled Model

	IC Exhaust-Heated 5650 Model	Optimal IC Exhaust-Heated 5650 Model
Number of Disks	2	2
Core Type	503A	503A
Effectiveness	0.975	0.975
Diameter of Each Disk (m)	3.5198	3.4688
Thickness of Each Disk (m)	0.1386	.1352
Mass of Each Disk (kg)	853.7	808.8
Rotational Speed (RPM)	1.74	1.85
Power Consumption (kW)	11.62	12.02
Total Radial Seal Leakage (% WA1)	3.22	2.06
Total Circumf. Seal Leakage (% WA1)	1.41	.89
Cold Side		
Pressure drop (%)	.17	.41
Heat-transfer area (m^2)	1467.9	1396.3
Free-face area (m^2)	1.351	1.318
Face area (m^2)	1.908	1.861
Hot Side		
Pressure drop (%)	3.11	3.08
Heat-transfer area (m^2)	4280.3	4049.5
Free-face area (m^2)	3.940	3.821
Face area (m^2)	5.565	5.397

6.0 Intercooled Design Options

There are three alternatives considered for the modification of the intercooled exhaust-heated 5650. Two of those options provide the optimal pressure ratio of 4.5 while one option keeps the original design pressure ratio. These options are:

1. run the modified engine at the existing pressure ratio;
2. design all-new turbomachinery; or
3. make no turbomachinery modifications: just decrease engine speed.

The first option compromises the optimum pressure ratio for economic comparison while the latter two options attain the optimal pressure ratio determined for the intercooled version.

6.1 Applicable Nomenclature and Base Engine Data

Nomenclature for both the compressor and turbine velocity diagrams use the notation found in Wilson [5]and are depicted in figure 6.1 and 6.2. The base-5650-compressor velocity-diagram data are presented in Table 6.1. Base-compressor dimensions are presented in Table 6.2.

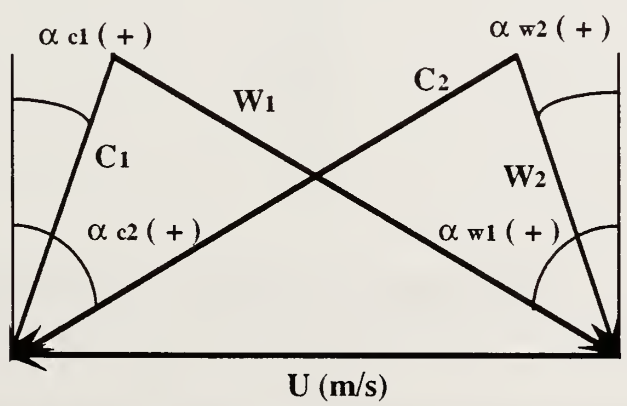


Figure 6.1 Compressor Velocity-Diagram Conventions [31]

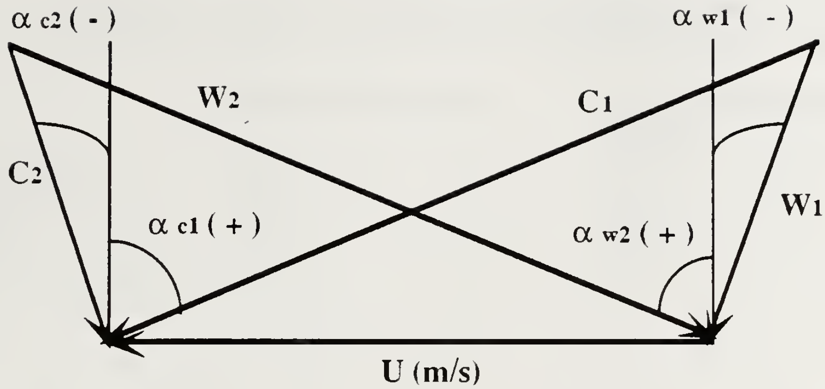


Figure 6.2 Turbine Velocity-Diagram Conventions [31]

Table 6.1 Base 5650 Compressor Velocity-Diagram Data

	Stage 1 Inlet Hub	Shroud	Exit	Stage 2 Inlet Hub	Shroud	Exit
u (m/s)	132	282	462	118	240	423
C (m/s)	151	151	335	87	87	294
α_c (°)	0	0	64	0	0	69
W (m/s)	201	319	216	146	225	182
α_w (°)	41	62	48	53	70	54

Table 6.2 Base 5650 Compressor Dimensions

	Stage 1	Stage 2
d_{1h} (mm)	193.1	171.4
d_{1s} (mm)	410.4	350.0
d_2 (mm)	673.5	616.8
b_2 (mm)	29.0	24.1
d_3 (mm)	743.5	675.9
b_3 (mm)	35.0	29.0
d_4 (mm)	1042.8	952.3
β_2 (°)	40	50
Z	26	26

6.2 Option 1- Run Engine At Original Pressure Ratio

Although this option may seem trivial , it should provide an economic alternative to the redesign of all turbomachinery. Table 5.7 and 5.8 previously presented the performance of this option and compared it to the intercooled base 5650 model developed from the predicted data provided by Solar. Table 5.7 is again presented below.

Table 6.3 Overall Performance Comparison
Intercooled Base Model vs. Intercooled Exhaust-Heated Model

	IC Base 5650 Model	IC Exhaust-Heated 5650 Model
Thermal Efficiency (%)	36.6	35.8
Power Output (kW)	3520	3257
Specific Power	.647	.599
Specific Fuel Consumption (kg/kW-hr)	.2312	.2938

The validity of this option will be proven and compared using an economic analysis model in section 9.0.

6.3 Option 2 - Redesign All Turbomachinery

A second alternative to achieve the optimal pressure ratio for the intercooled exhaust-heated 5650 involves redesigning all of the turbomachinery. The efficiency of the redesigned turbomachinery will be maximized at the low-pressure-ratio design point and presumably will result in better overall engine performance. Option 1 for the intercooled exhaust-heated engine required simply running the modified engine at its original design pressure ratio. This avoided the redesign of any turbomachinery Both compressor stages and both turbines must be redesigned in option 2.

The capital cost of more turbomachinery modifications will be weighed against savings from performance improvements on a life-cycle basis in section 9.0. In an effort to keep the flowpath and overall size of the engine as near to the base 5650 design as possible, the mass-flow rate and turbine-inlet temperature were held constant. The compressor rotational speed was allowed to vary from the base 5650 design speed but was constrained by the existing dimensions of the compressor turbine in order to simplify the alteration of this turbine. The preliminary design of the turbines was accomplished using Tampe's TURBINE computer program. The centrifugal-compressor preliminary design was completed based on procedures in Wilson [5] and lectures by Professor A. D. Carmichael

in MIT's Thermal Power Systems course. A second iteration would incorporate a backswept impeller of about 45 degrees.

Backswept vanes have several advantages over radial vanes in that relative tip velocities increase while the absolute velocity of the fluid decreases. These velocity changes result in less stringent diffusion requirements in both the impeller and diffuser which tend to increase the efficiency of these components. Backswept vanes also provide the compressor with a wider operating range of air-flow for a given rotational speed, simplifying the match of the compressor to its driving turbine. One disadvantage of backswept vanes is the reduction of work-absorbing capacity of the rotor resulting in lower temperature rises as compared to a similar radial-vaned impeller. This effect is countered by the increased efficiency of the components [22].

The centrifugal-compressor preliminary design followed the same design constraints as Nahatis [31]. The final design assumptions listed in Table 6.4 were compiled by him after consulting numerous references [32,33,34,35,36].

Table 6.4 Centrifugal-Compressor Design Assumptions

Overall	
Specific speed $N_S \equiv \frac{RPM}{60}\left(\frac{Q^{.5}}{(\Delta h)^{.75}}\right)$	~0.6
Slip factor	0.9
Isentropic impeller efficiency	92.0 %
Inlet axial velocity	uniform from hub to shroud
Inlet swirl	none
Impeller	
Relative Mach number at shroud inlet	minimize
Hub-to-tip ratio at inlet	0.28
Blade exit angle	0 °
Vaneless Diffuser	
Radius * tangential velocity ($R*C_\theta$)	constant
Mach number at exit	0.8
Pressure loss distribution	equal among diffuser components
Diffuser width	equal to impeller exit blade width
Vaned Diffuser	
Incidence	0 °
Flow entrance angle	vaneless diffuser exit angle
Flow exit velocity	1/3 of entrance velocity

Due to the iterative nature of centrifugal-compressor design, a computer program was developed to execute the calculations. To maintain continuity between intercooled and non-intercooled versions of this cycle which were completed under the same DOE contract, the same design constraints were met. For the intercooled cycle, the interstage pressure and temperature data from the optimized cycle results were used as inputs for the centrifugal-compressor design. Rotational speed was chosen in order to limit modifications to the compressor turbine. Loading coefficient was chosen while Pressure ratio and polytropic efficiency of the compressor stages were taken from the intercooled model. All other parameters were calculated.

The design parameters shown in Table 6.5 were the result of optimizing the intercooled-compressor design using the present dimensions of the compressor turbine as a constraint in order to avoid unnecessary changes to the casing and blade dimensions there.

Table 6.5 Intercooled Compressor Design Parameters

	Stage 1	Stage 2
Speed (RPM)	12500	12500
Pressure Ratio	2.26	2.04
Specific Speed	0.16	0.11
Polytropic Efficiency (%)	87.0	82.0
ϕ_1	0.47	0.38
ϕ_2	0.62	0.49
ψ	-0.9	-0.9

The results from the program, the final compressor performance and dimensions, are compared with the base 5650 parameters in Table 6.6. The velocity-diagram data for each stage are contained in Table 6.7 and a schematic of the two stages is shown in figures 6.3 and 6.4.

Table 6.6 Comparison of Redesigned Compressor Geometry Intercooled Exhaust-Heated Cycle

	Stage 1		Stage 2	
	Base 5650	Redesign	Base 5650	Redesign
d_{1h} (mm)	193.1	134.4	171.4	136.8
d_{1s} (mm)	410.4	404.9	350.0	323.5
d_2 (mm)	673.5	480.1	616.8	488.4
b_2 (mm)	29.0	46.2	24.1	28.5
d_3 (mm)	743.5	543.8	675.9	523.8
b_3 (mm)	35.0	46.2	29.0	28.5
d_4 (mm)	1042.8	802.6	952.3	726.5
β_2 (°)	40	0[4]	50	0[4]
Z	26	20	26	20

[4] A second design iteration would incorporate an impeller backswept about 45 degrees.

Table 6.7 Redesigned Compressor Velocity-Diagram Data Intercooled Exhaust-Heated Cycle

	Stage 1 Inlet Hub	Shroud	Exit	Stage 2 Inlet Hub	Shroud	Exit
u (m/s)	88	265	314	90	212	320
C (m/s)	148	148	343	123	123	327
α_c (°)	0	0	55	0	0	62
W (m/s)	172	303	197	152	245	159
α_w (°)	31	61	9	36	60	12

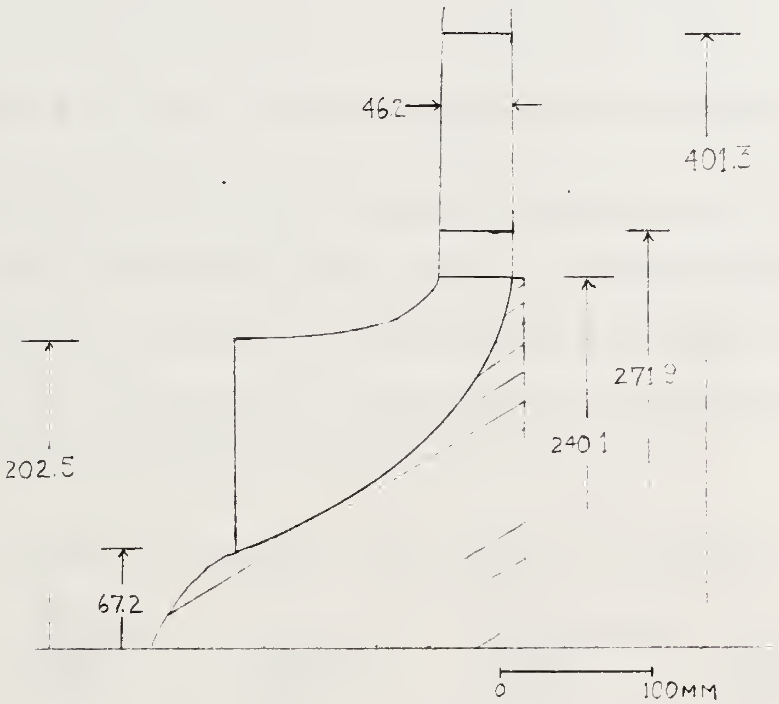


Figure 6.3 Stage 1 Schematic (Intercooled Compressor)

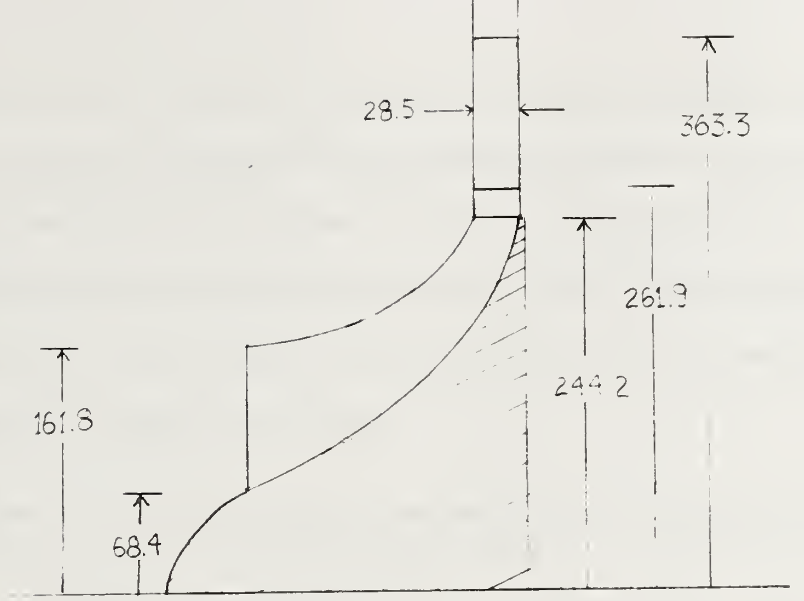


Figure 6.4 Stage 2 Schematic (Intercooled Compressor)

The design-point performance of each centrifugal compressor stage was verified using Dallenbach's performance-prediction method which was encoded by Nahatis [26]. The new stage geometry was entered into the base 5650 data-match computer program. The resulting efficiency and pressure-ratio prediction for the new geometry compared reasonably well with design intent (see Table 5.5).

Table 6.8 Intercooled Compressor Design Intent vs. Prediction

	Efficiency Design	Predicted	Pressure Ratio Design	Predicted
Stage 1	87.0	89.6	2.21	2.11
Stage 2	82.0	85.3	2.04	2.16

The preliminary design of the turbines was completed in a similar manner as the turbine designs performed by Nahatis [31]. The preliminary design of the turbine blading was carried out with the help of Tampe's TURBINE computer program [16]. The program uses preliminary constant-hub-diameter design procedures outlined in Wilson [5] with the final designs meeting criteria advocated by Wilson [37] Data for the original turbine designs were calculated knowing the rotor and stator dimensions and assuming a reaction

of .6. The data given in Table 6.9 and shown in the velocity diagrams in figure 6.5 are provided for comparison purposes with the redesigned turbines at the optimized pressure ratio. The constant-hub-diameter geometry for the gas-producer turbine is compared with the base 5650 cylindrical annulus design in Table 6.10. The chosen compressor speed of 12,500 rpm resulted from seeking to minimize the changes to the gas-producer turbine when running at its modified pressure ratio.

Table 6.9 Mean-Diameter Turbine Velocity-Diagram Data For Baseline 5650

	Gas-Producer Turbine		Power Turbine	
	Inlet	Exit	Inlet	Exit
Rn	---	0.6	---	.6
ψ	---	2.0	---	1.6
ϕ	---	0.79	---	0.73
C (m/s)	580	358	453	268
α_c (°)	61	-37	58	-16
W (m/s)	320	644	244	509
α_w (°)	-27	64	-16	63

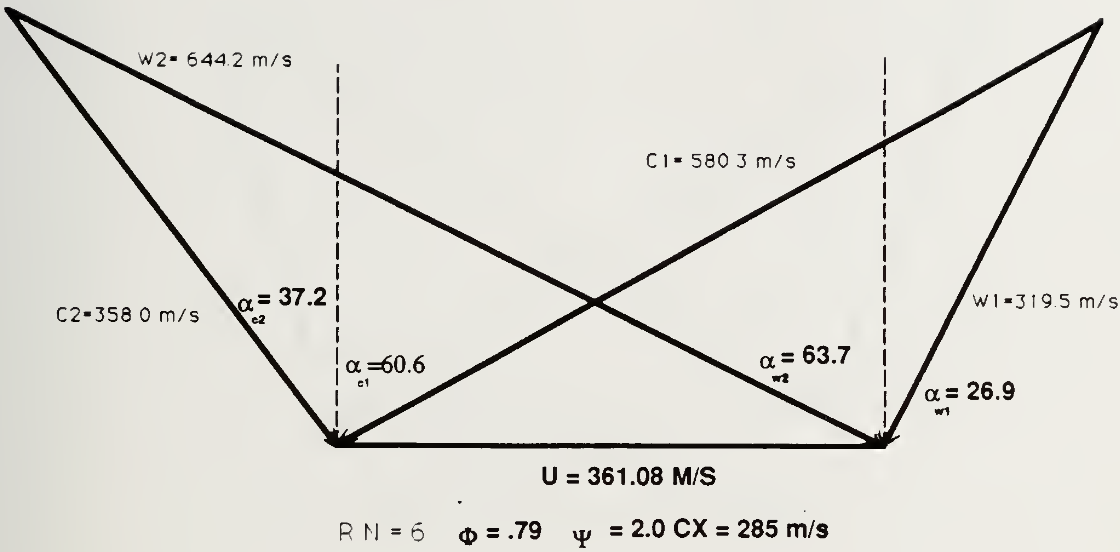
Table 6.10 Gas-Producer Turbine Geometry For Redesigned Intercooled Compressor Option

	Base 5650		Redesign	
	Vane	Blade	Vane	Blade
d_h (mm)	463.6	463.6	463.7	463.7
d_t (mm)	589.3	589.3	535.3	555.6
λ	0.787	0.787	.866	0.835
A_n (m^2)	0.1040	0.1040	0.0562	0.0735

The geometry for the power turbine is shown with the base 5650 dimensions in Table 6.11. The power-turbine speed remains the same as the base engine but the pressure ratio has now been changed due to operation at the chosen optimal pressure ratio. The velocity diagrams for the redesigned turbines are shown in figure 6.6. A schematic of the redesigned turbines is presented in figure 6.7.

SOLAR 5650 TURBINE VELOCITY DIAGRAMS

COMPRESSOR TURBINE



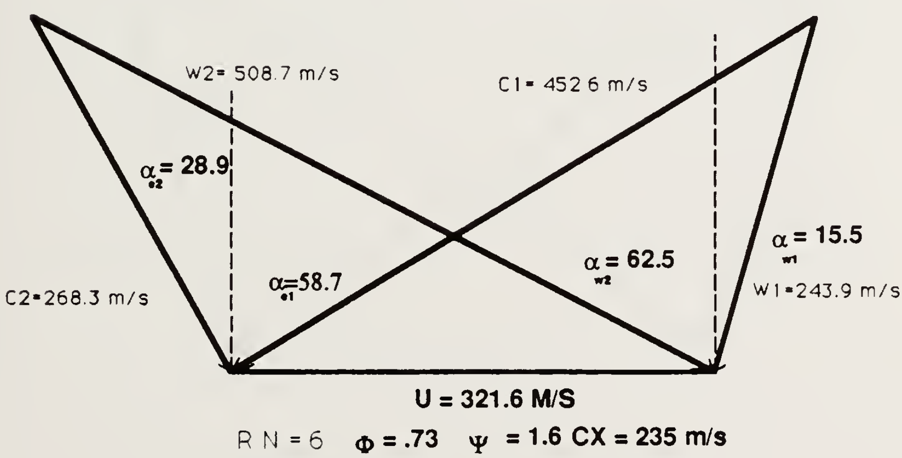
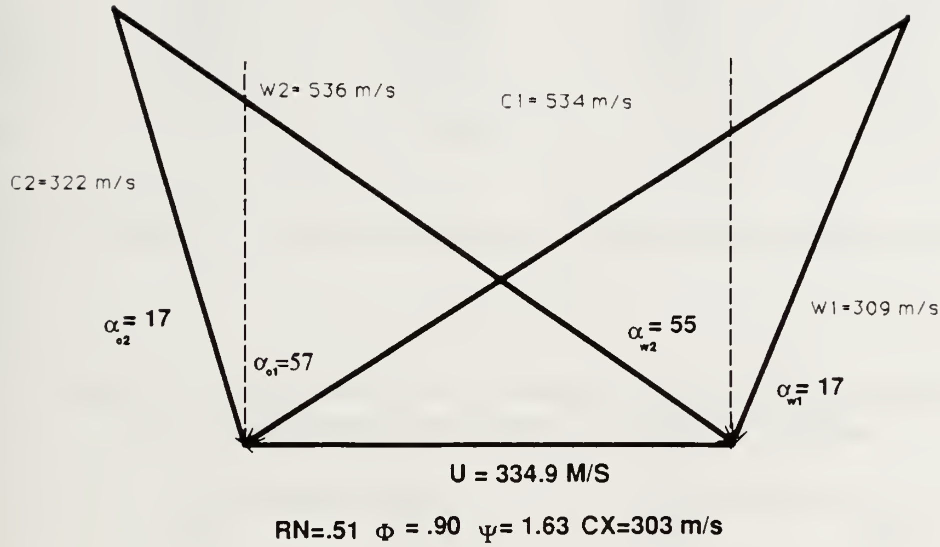


Figure 6.5 Baseline 5650 Turbine Velocity Diagrams (Mean Diameter)

REDESIGNED TURBINE VELOCITY DIAGRAMS

COMPRESSOR TURBINE



POWER TURBINE

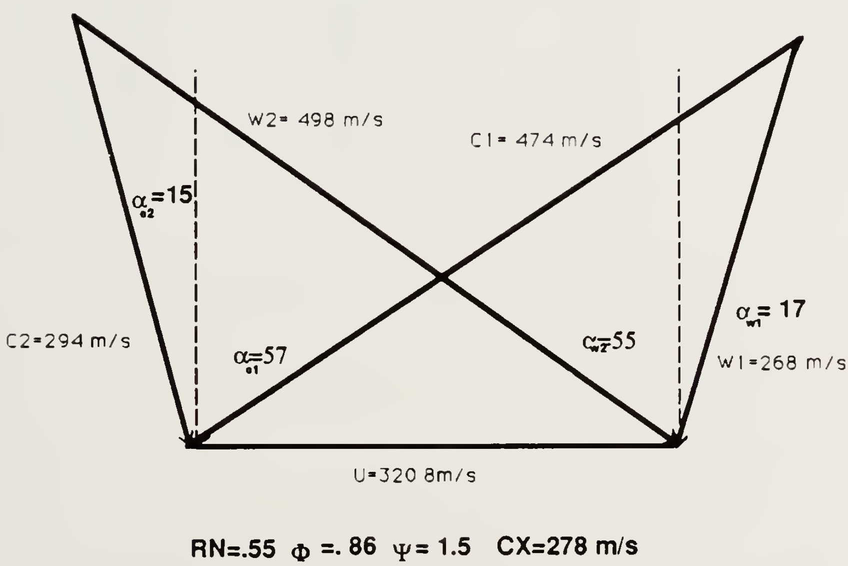


Figure 6.6 Velocity Diagrams for Redesigned Turbines (Mean Diameter)

Table 6.11 Power-Turbine Geometry For Redesigned Intercooled Compressor Option

	Base 5650 Vane	Blade	Redesign Vane	Blade
d_h (mm)	468.6	468.6	500.0	500.0
d_t (mm)	688.2	688.2	614.7	644.7
λ	0.681	0.681	0.813	0.776
A_n (m^2)	0.1996	0.1996	0.1004	0.1301

The mean-diameter velocity-diagram data for the redesigned turbines are shown in Table 6.12.

Table 6.12 Mean-Diameter Turbine Velocity-Diagram Data For Redesigned Intercooled Compressor Option

	Gas-Producer Turbine Inlet	Exit	Power Turbine Inlet	Exit
Rn	---	0.51	---	.55
ψ	---	1.63	---	1.49
ϕ	0.86	0.90	0.80	0.86
C (m/s)	534	322	474	294
α_c (°)	57	-17	57	-15
W (m/s)	309	536	268	498
α_w (°)	-22	55	-17	55

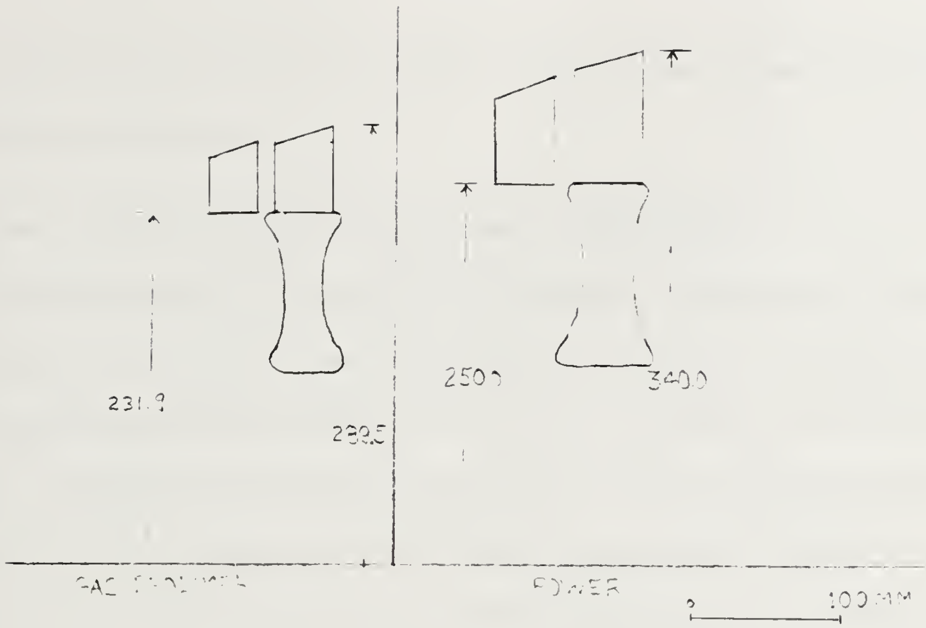


Figure. 6.7 Turbine Schematic

The design-intent efficiency of both turbines was verified using three turbine efficiency-prediction techniques: Wilson's exact, the trimmed binomial method, and a method developed at General Electric [5,38,39]. The results shown in Table 6.13 agree moderately well with the design-intent polytropic efficiency.

Table 6.13 Design Intent vs. Predicted Efficiency Intercooled Exhaust Heated Cycle

	Design Intent	Wilson's Exact	Trim Binom.	GE
Gas-Producer Turbine	88.7	87.9	90.1	89.8
Power Turbine	87.7	89.5	91.7	91.6

The preliminary design of the compressor and both turbines has been completed for the intercooled exhaust-heated engines. The blading performance and efficiencies have been estimated to be better than design intent. To be conservative, however, the intercooled exhaust-heated 5650-engine model use the design-intent efficiencies. The last option which

will be analyzed involves running the modified engine at reduced speed and involves no turbomachinery modifications.

6.4 Option 3 - Run Existing Turbomachinery Off-Design

The off-design running of the intercooled, exhaust-heated, coal-burning 5650 with no turbomachinery modifications is the last option considered to arrive at the optimal pressure ratio for high thermal efficiency and specific power. This exercise is divided into two major tasks: determining the off-design performance of the base 5650 turbomachinery and predicting the off-design characteristics of the rotary regenerator sized in Table 5.5.

The off-design performance of the base 5650 turbomachinery was determined from actual test data shown in figures 6.8, 6.9, and 6.10 [21]. The data were extrapolated down to lower pressure ratios based on the assumption that the slopes stayed constant.

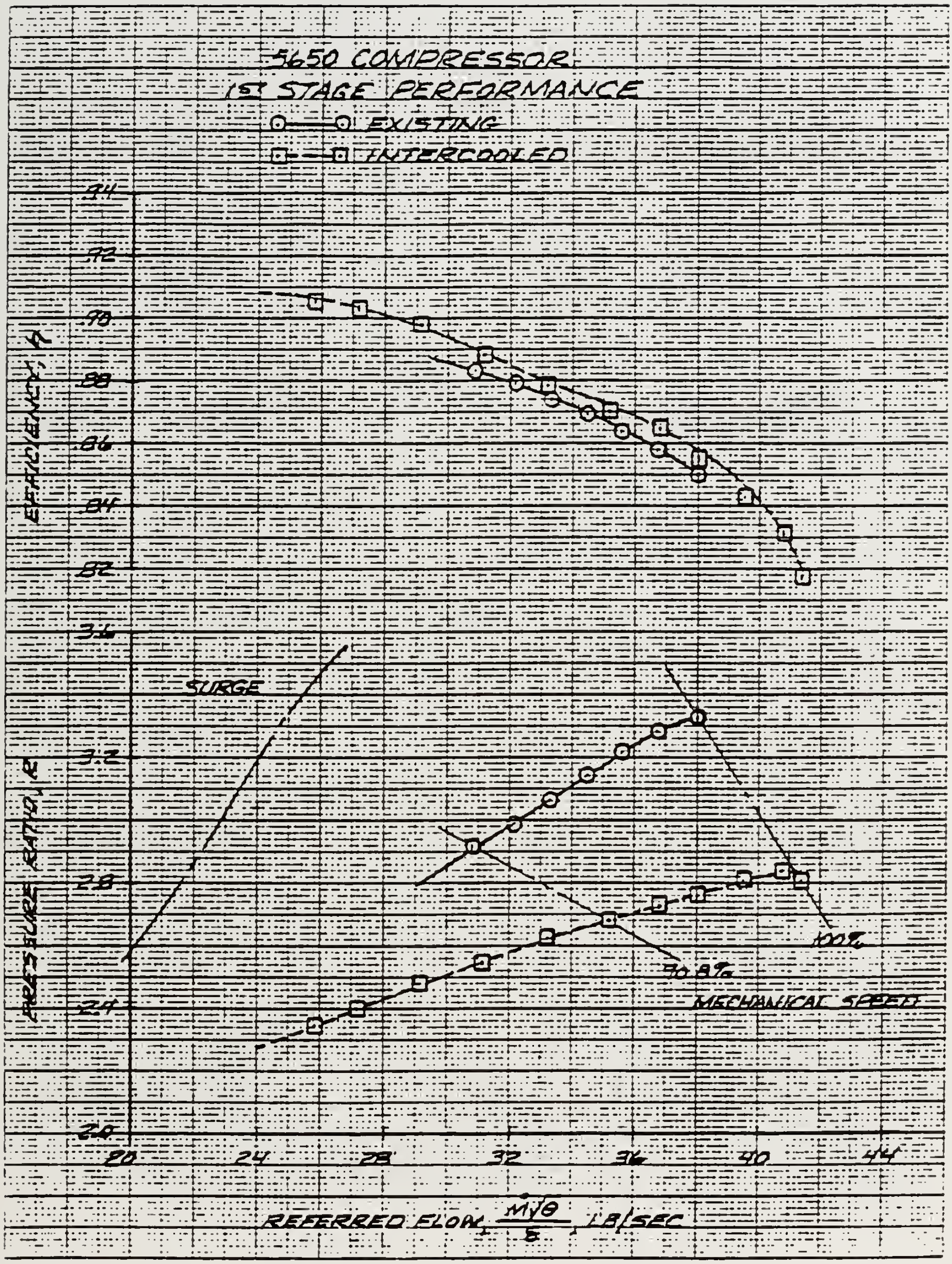


Figure 6.8 Intercooled Solar 5650 Compressor

First Stage Operating Line [21]

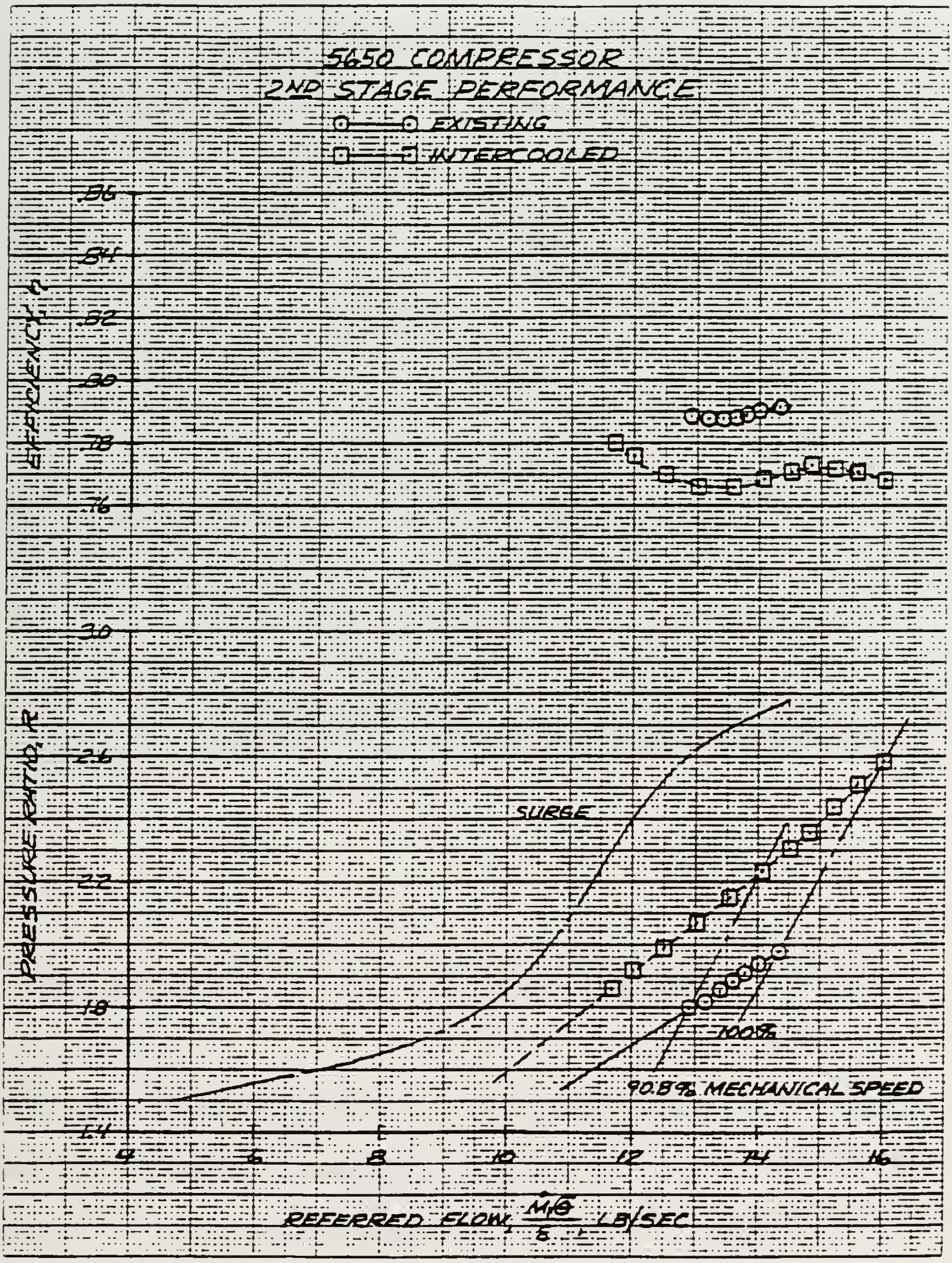


Figure 6.9 Intercooled Solar 5650 Compressor Second Stage Operating Line [21]

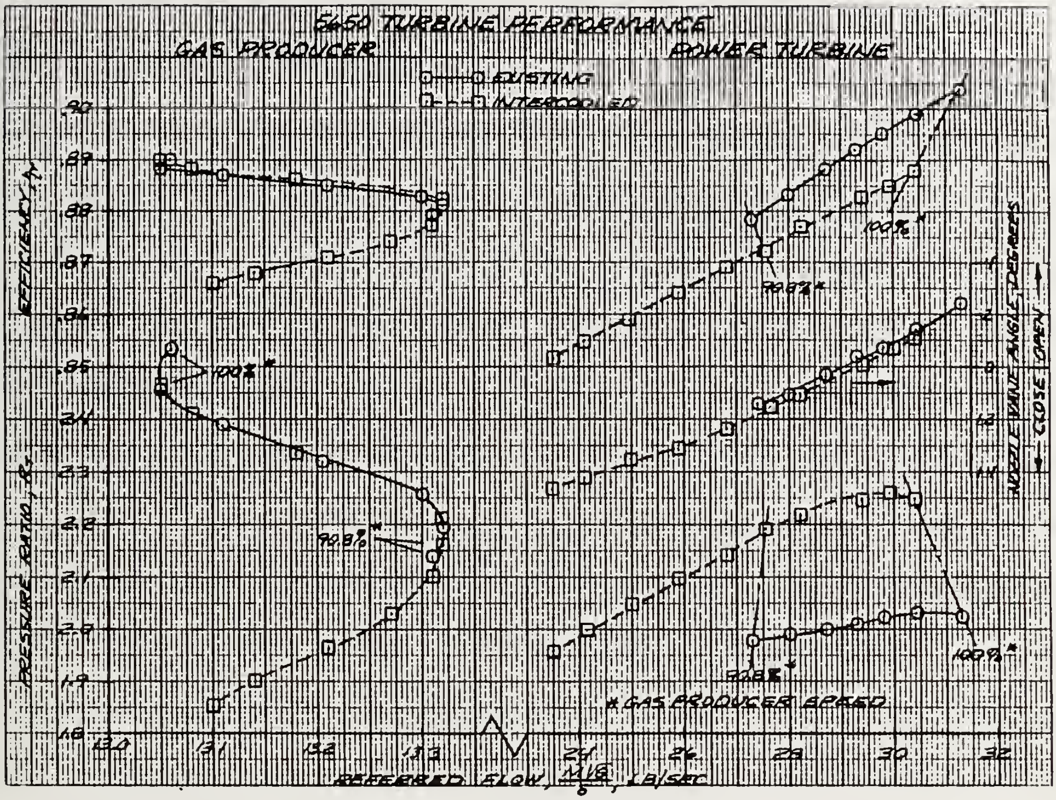


Figure 6.10 Solar 5650 Turbine Operating Line [21]

The off-design-point characteristics of the rotary regenerator were estimated using techniques documented in Hagler [19] and Frenkel [15]. The variation of effectiveness with mass flow through the regenerator is assumed to be linear based on analysis by Frenkel [15].

The off-design-point-engine performance was calculated for a range of pressures using a computer program which merged elements from Tampe's CYCLE program and Frenkel's regenerator off-design calculations . The program requires interactive input from the test

data in figures 6.8 , 6.9 and 6.10 for each pressure ratio. The dimensions of the rotary regenerator, sized at the design-point pressure ratio (Table 5.5), are held constant.

The thermal-efficiency-versus-specific-power characteristic for the off-design-point running of the intercooled exhaust-heated 5650 is shown in figure 6.10. A relatively high thermal efficiency was reached but the power output at such low pressure ratios is too small for the engine to be economically justifiable. A comparison of thermal-efficiency curves with design and off-design points reveals that the off-design-point thermal efficiencies follow design-point values closely at lower and upper extremes (see figure 6.11). These results depict the improvement in part-load performance that intercooling and regeneration have on the cycle. These results may then be compared with the part-load performance comparison that was performed by Nahatis [31] who investigated the non-intercooled cycle. Whether the increased initial capital expense of the other options is worth the performance improvement is determined in section 9.0. An overall performance comparison of the optimal-exhaust-heated 5650 and the off-design-point-exhaust-heated 5650 at a pressure ratio of 4.0 is shown in Table 6.14.

Table 6.14 Overall Performance Comparison
Optimal vs. Reduced Speed (Non-intercooled)

	Optimal Design	Reduced-Speed Operation
Thermal Efficiency (%)	38.5	36.8
Power Output (kW)	2490	1479
Specific Power	0.500	0.450
Specific Fuel Consumption (kg/kW-hr)	0.2731	0.2859
Pressure Ratio	4	4
Mass Flow (kg/s)	17.22	11.34

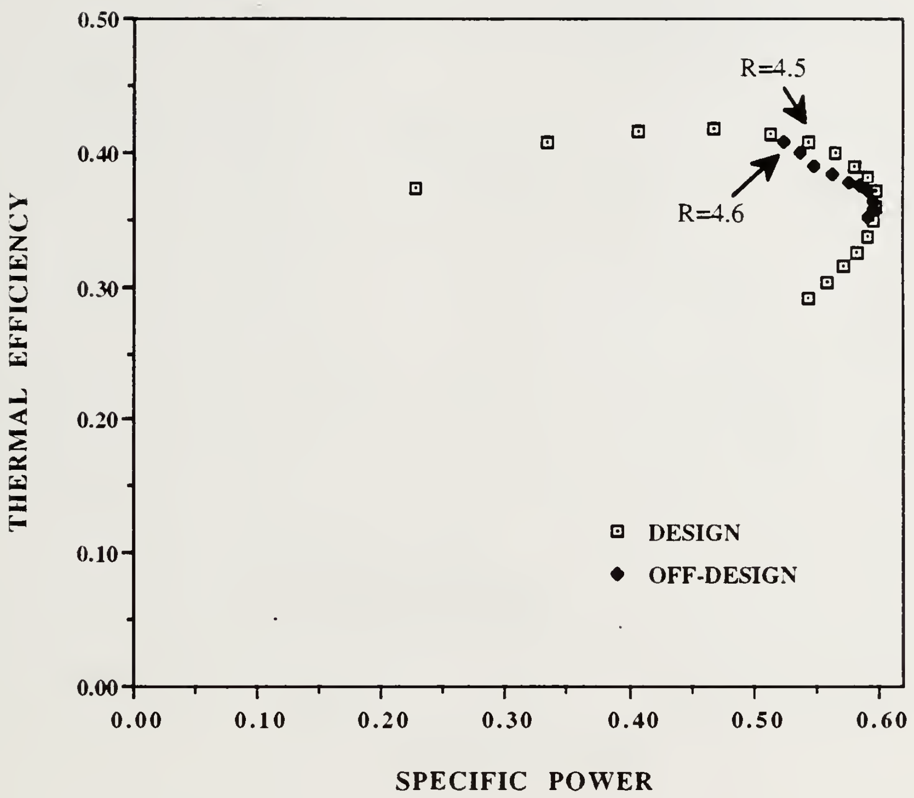


Figure 6.11 Thermal Efficiency Comparison (Intercooled Exhaust-Heated Cycle)

Table 6.15 provides a performance comparison between options 2 and 3 for the intercooled exhaust-heated modification. Although the reduced-speed (off-design) thermal efficiency matches that of the turbomachinery redesign option, this is achieved at a great sacrifice in net power.

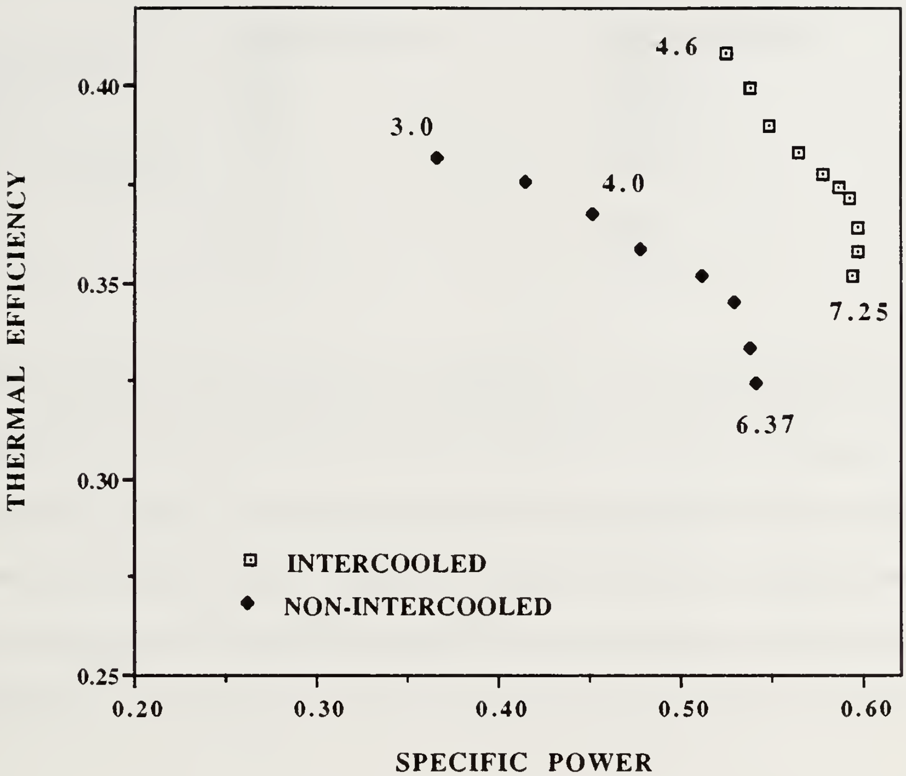


Figure 6.12 Reduced-Speed Thermal Efficiency vs. Specific Power

Figure 6.11 shows a comparison between the optimal design-point and the reduced-speed (off-design) performance of the intercooled exhaust-heated cycle. The reduced-speed performance curve of the intercooled model operates much closer to its optimal design points as compared to the non-intercooled model shown in figure 6.12.

Table 6.15 Overall Performance Comparison
Optimal vs. Reduced-Speed (Intercooled Exhaust-Heated Cycle)

	Optimal Design	Reduced-Speed
Thermal Efficiency (%)	40.8	40.9
Power Output (kW)	2961	1875
Specific Power	.545	.525
Specific Fuel Consumption (kg/kW-hr)	.2577	.2571
Pressure Ratio	4.5	4.6
Mass Flow (kg/s)	18.77	12.34

The thermal efficiency and specific fuel consumption are quite comparable but the net power output of the off-design models is approximately 37% less than the net power of the optimal design-point model. The reduced mass flow at lower pressure ratios is the primary reason for the power output deficit. While the thermal efficiency, specific power and specific fuel consumption of the reduced-speed option look attractive, the absolute power output is significantly below the level reached when new turbomachinery is designed.

7.0 Increased Turbine-Inlet Temperature

The intercooled exhaust-heated 5650 model was run at the optimal pressure ratio with increased turbine-inlet temperature to determine the potential benefit to the overall cycle performance. Solar maintains that more effective gas-producer turbine-blade cooling would allow the turbine-inlet temperature of the engine to climb from 1241 K to 1339 K. Although Solar does not mention any other changes, for a conservative estimate the cooling flow to the gas-producer turbine was increased from 2.5% to 3.5 % and power-turbine cooling was increased from 0.8% to 1.5 %. The resulting overall performance is compared to the optimal pressure-ratio, intercooled exhaust-heated 5650 model in Table 7.1.

Table 7.1 1339 T.I.T. Cycle Comparison
Intercooled Exhaust-Heated Cycle

	Optimal Exhaust-Heated	1339 K T.I.T.
Thermal Efficiency (%)	40.8	43.3
Power Output (kW)	2961	3377
Specific Power	.544	0.621
Specific Fuel Consumption (kg/kW-hr)	0.2577	0.2427

The potential performance benefit from this cycle is enormous. Both power output and efficiency increase while specific fuel consumption decreases. Although this is true, there may also be some potential disadvantages to this cycle. The life of some of the uncooled engine parts may be compromised and the increased coal-firing temperature could lead to a significant rise in the stickiness of the coal ash which would have an adverse effect on the regenerator operation. Nevertheless, the prospect of running a more-efficient and-powerful cycle is noteworthy.

The increase in performance of the intercooled exhaust-heated cycle is very similar to the increase obtained with the non-intercooled exhaust-heated cycle with TIT raised to 1339 K [26]. A composite plot of the performance of the various intercooled options and cycle modifications for each exhaust-heated engine model compared with the base intercooled

5650 engine is shown in figure 7.1. Three options show better efficiency at less absolute power output than the respective base 5650 engine. The intercooled exhaust-heated engine which keeps the same pressure ratio as the original base engine exhibits less power and efficiency. The increased TIT cycle modification for the optimized, intercooled exhaust-heated model has significantly higher efficiency and slightly less power. The data will be compared on a life-cycle-cost basis in a following section.

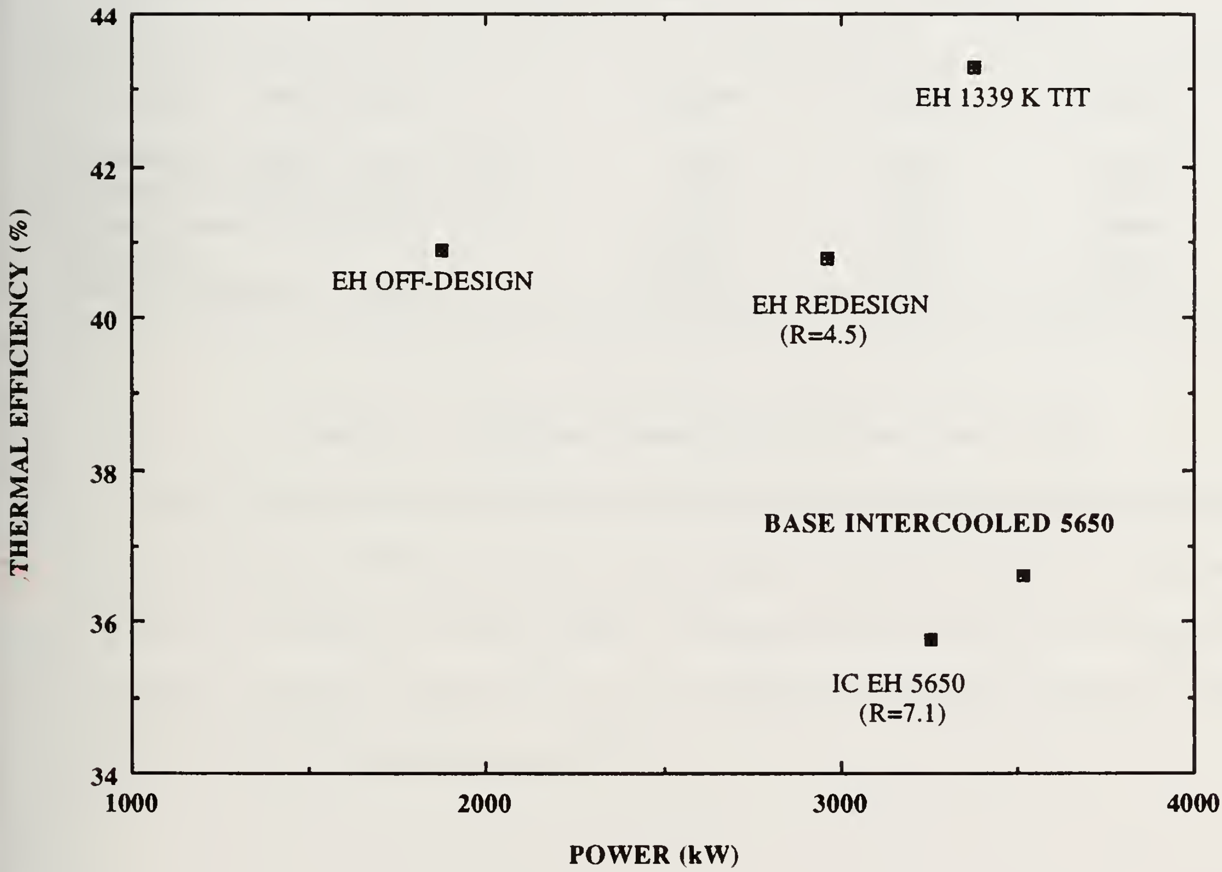


Figure 7.1 Design Performance of All Intercooled Options

8.0 REGENERATOR MATRIX SIZING EFFECTS

Cycle analysis runs were completed for the three cores with characteristics shown below in table 8.1. Revised pressure and mass loss data were incorporated in the performance calculations and the effectiveness of the cores was chosen in all runs to be .975. Each run was for the intercooled exhaust -heated cycle with optimized pressure ratio. Sizing the regenerators follows the Kays and London NTU method in the form used by Wilson [5]. The program developed by Tampe [16] also calculates the mass flow leakage across the heat-exchanger seals using the equations developed by Hagler [19].

Core No. (Stanford)	505A	503A	504A
Passage Count, No./in^2	526	1008	2215
Hydraulic diameter, μm	753	511	327
Area density, m2/m3	4216	5551	7864
Porosity	0.794	0.708	0.644
Solid density, kg/m^3	2259	2259	2259

Table 8.1 Surface Geometry For Three Cores [40]

It may be practical for this cycle to use regenerator cores with larger hydraulic diameters in order to reduce fouling or decrease the cleaning or replacement intervals of the cores. Figures 8.1 through 8.4 display changes in regenerator mass, cycle power output, disc diameter, and disc thickness for cores of different hydraulic diameters, holding effectiveness for all program runs at .975.

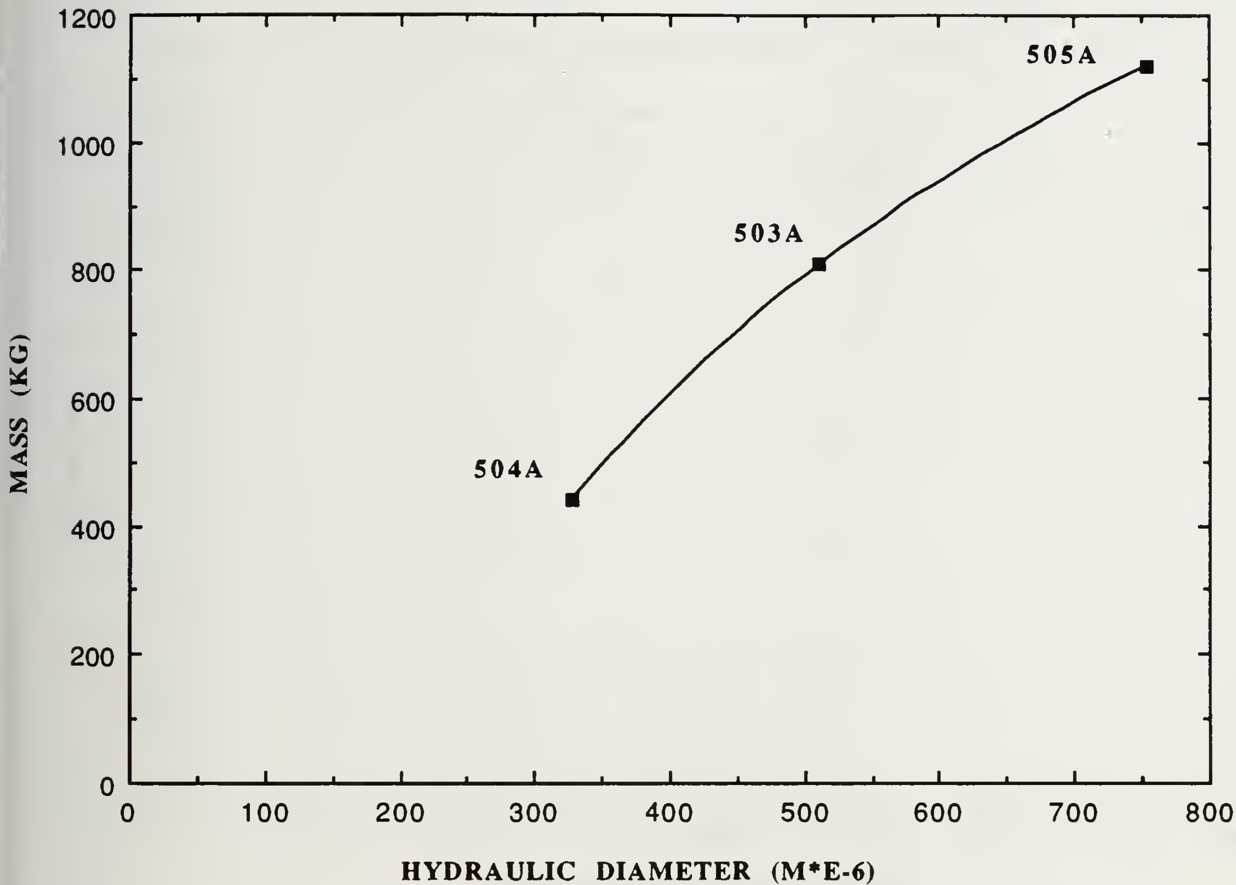


Figure 8.1 Regenerator Disc Mass Versus Hydraulic Diameter

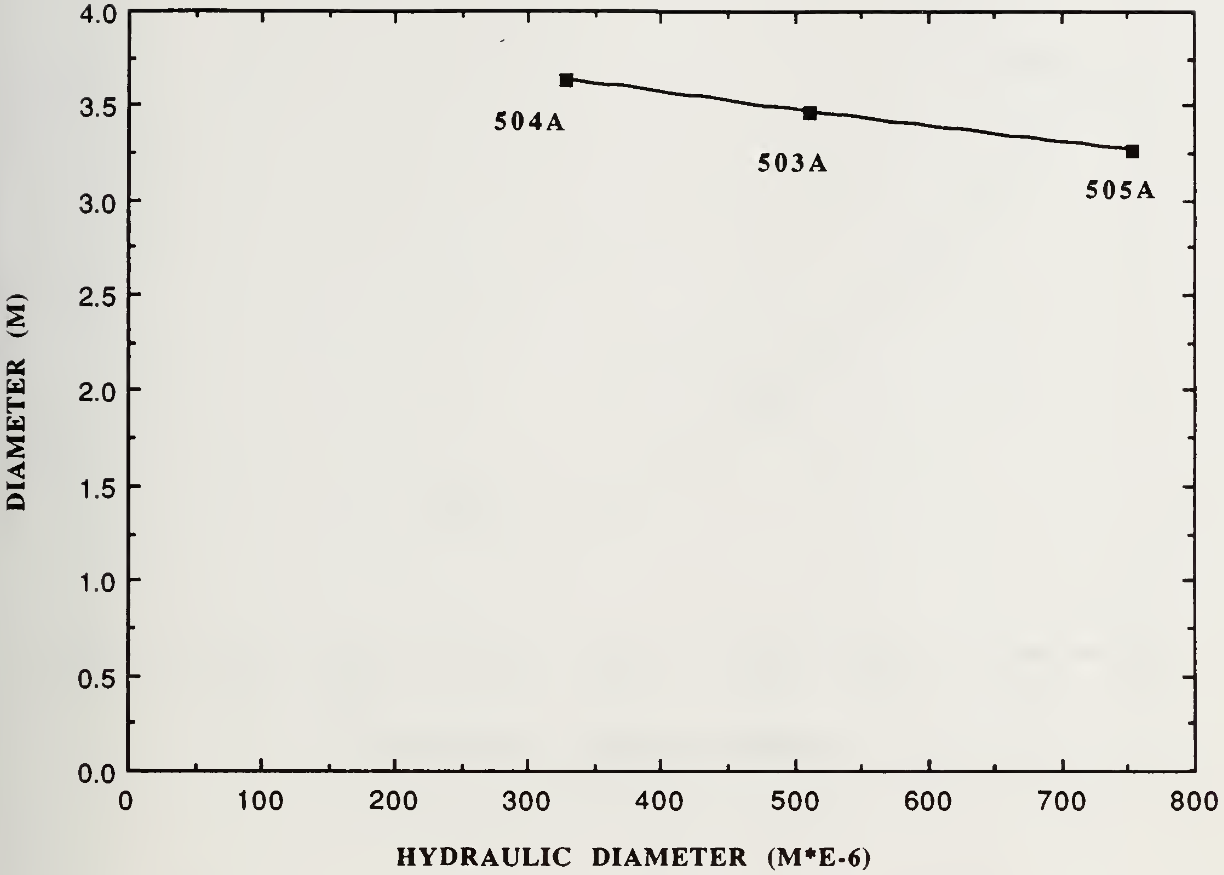


Figure 8.2 Regenerator Disc Diameter Versus Hydraulic Diameter

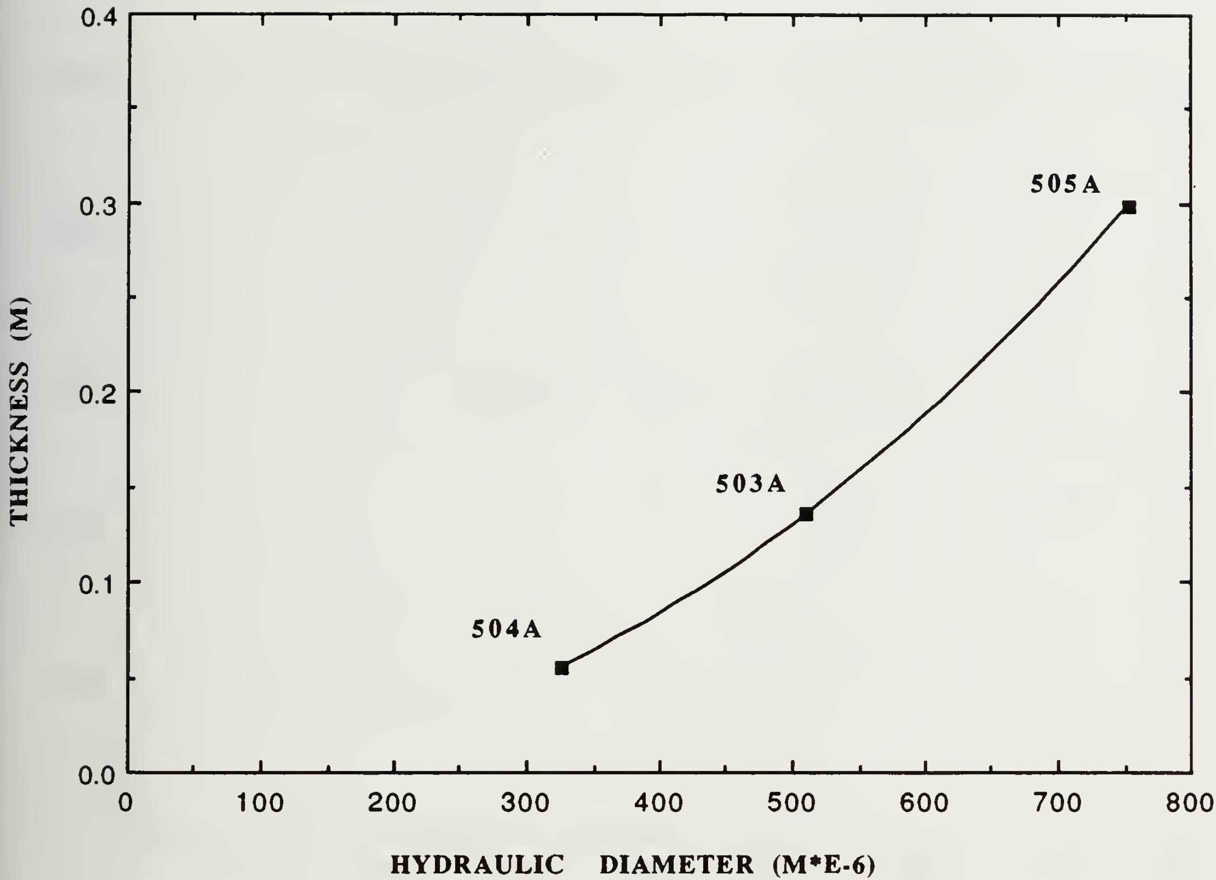


Figure 8.3 Regenerator Disc Thickness Versus Hydraulic Diameter

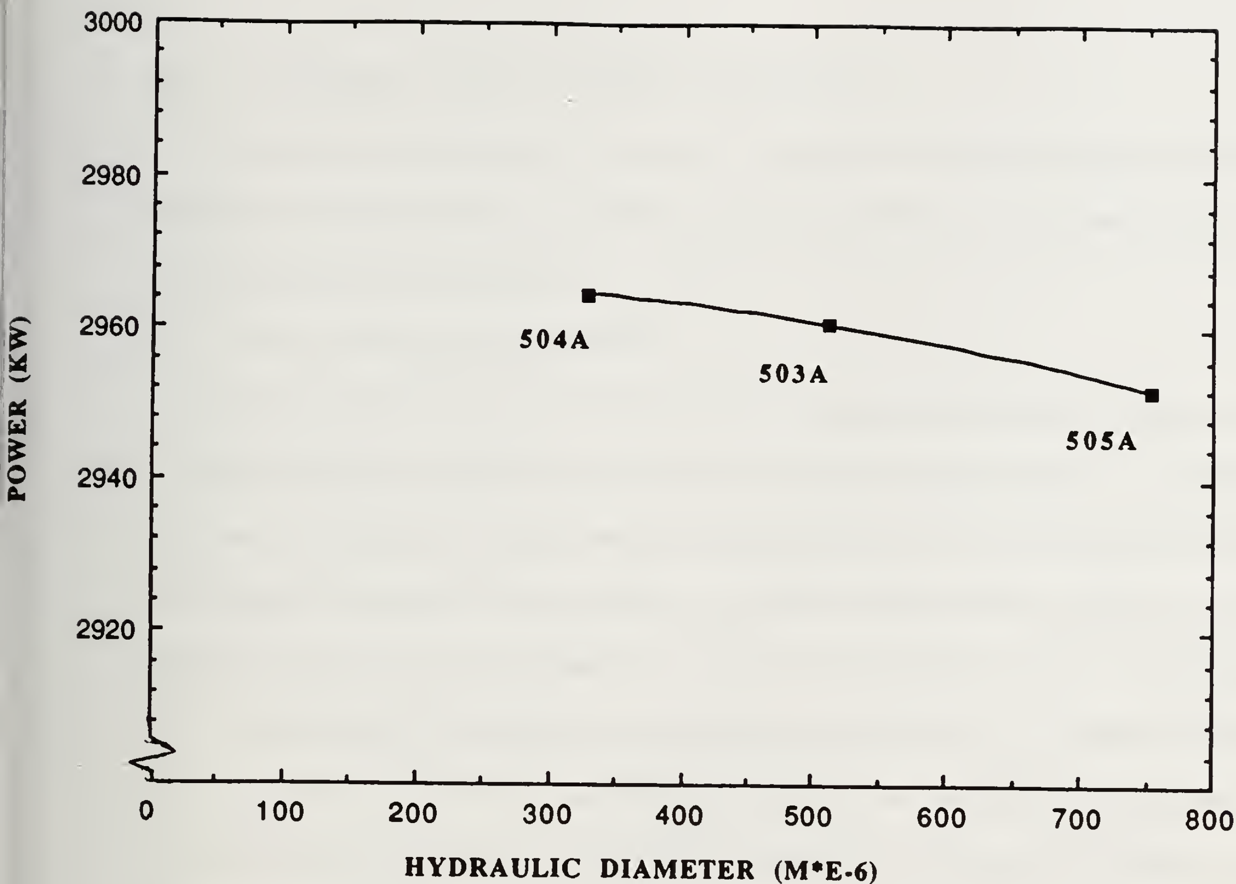


Figure 8.4 Cycle Output Power Versus Hydraulic Diameter

From the graphical results it is apparent that thickness and mass are affected the most when a core of larger hydraulic diameter is chosen for the cycle. Power output and disc diameter changes are small in comparison. Cycle power output varied for the different cores due to changes in mass and pressure losses as well as power taken from the cycle to drive the regenerators. Data from these runs is contained in Table 8.2.

Table 8.2 Data From Varied Core Runs

Core Type	Hyd. Diam.	Mass/Disc	Power (kW)	Diam. (M)	Thick. (M)
504 A	327μm	441.2 Kg	2951.9	3.64	0.055
503 A	511μm	808.6 Kg	2960.7	3.47	0.135
505 A	753μm	1120.5 Kg	2964.4	3.27	0.298

9.0 ECONOMIC COST-BENEFIT ANALYSIS

The goal of this section is to make an accurate life-cycle-cost assessment of the intercooled exhaust-heated cycle options in order to determine which project performs the best from an economic standpoint

9.1 The Life-Cycle-Cost Method

The life-cycle-cost model used in this analysis was developed by R. B. Spector [41] to evaluate the relative merits of varying types of industrial gas turbines. The following elements are considered in the model: initial investment cost, cost of financing, variations in equipment availability, cost of fuel, cost of fuel treatment and/or preparation, direct operating labor costs and spare parts for preventive and corrective actions. These elements are contained in the three terms which comprise the production cost: annual investment cost, annual fuel cost, and annual maintenance cost. The life-cycle-cost equation (9.1) calculates the average present value cost per kilowatt-hour of electricity generated over the life of the unit.

$$C_T = I\,\frac{\left[\dfrac{i}{1-(i+1)^{-n}}\right]}{(A)(kW)(8760)(G)} + \frac{F}{(293)(E)} + \frac{M}{kW} \qquad (9.1)$$

here,

I- initial capital cost of the equipment ($)
i- interest rate
n- number of payment periods
A- availability (number of hours engine operates/number of hours needed)
kW- number of kilowatts of electricity produced (kW)
G- efficiency of the associated electrical alternator
F- fuel cost ($/MBTU, HHV basis)
E- thermal efficiency
M- maintenance cost ($/hr)

The accuracy of the method is adequate for the purpose of evaluating the relative costs of the different options but Spector does not advocate its use to calculate absolute costs.

9.2 The Life-Cycle Equation Unknowns

The unknowns in the life-cycle equation (7.1) can be divided into two categories: those terms that vary with engine configuration and those that do not. Engine configurations have been categorized as either exhaust or direct-heated. The fuel cost, maintenance cost, availability, interest rate, generator efficiency, and payment periods do not change with engine configuration considered. The values of these invariants shown in Table 9.1 were arrived at through a comprehensive search of the literature [3,4,41,42,43]. The base 5650 is an entirely different engine and therefore requires its own set of constants also included in Table 7.1. Maintenance costs for the exhaust-heated cycles were chosen to be twice that of a "simple-cycle" gas turbine or equivalent to the maintenance costs of a diesel engine [44].

Table 9.1 Life-Cycle Calculation Constants

	Exhaust-Heated 5650	Base 5650
Fuel Cost ($/MBTU)	(Coal) 1.86	(Natural Gas) 2.97
Maintenance ($/kWhr)	.01	.005
Availability	0.95	0.98
Interest Rate	0.075	0.075
Periods	20	20
Generator Efficiency	0.98	0.98

Although optimistic, the cost of coal in this study is $1.86 /MBTU [44]. The actual cost will depend on how far from the coal source the plant is located and what treatments must be added to the coal to control the products of combustion. The $2.97 /MBTU fuel cost for the base 5650 is the current projected price of natural gas [44]. The maintenance cost of the coal-burning engine is chosen to be double the average cost for industrial gas turbines because the regenerator and its associated seals and the combustor and cleanup system will most likely require more frequent servicing than a simple-cycle gas turbine requires (this may, however, be too conservative) The balance of the terms in the life-cycle equation (9.1): initial capital cost, kilowatts, and thermal efficiency, vary with engine configuration.

The initial cost of the Solar 5650 with the exhaust-heated modifications is difficult to estimate. The cost of the base 5650 unit is not well established because Solar has leased them, not sold them, and then only to a limited number of pilot sites. A "rough" price for the Solar 5650 without the recuperator but including installation and generator cost was obtained from Solar. The price of an intercooler for the two-stage centrifugal compressor was obtained from Karstensen [21] The cost of the atmospheric-pressure slagging combustor, fuel system and extra ducting was simply estimated. The regenerator core cost was arrived at through conversations with a manufacturer and the price of the turbomachinery modifications was scaled using sample engine data supplied by a gas-turbine engine manufacturer. The total initial cost of each option examined is broken down into components in Table 9.2.

Table 9.2 **Initial Production Costs** (000 omitted)
Intercooled Exhaust-Heated Models

	IC 5650	IC EH 5650	IC EH Redesign	IC EH Off-Design	IC EH 1339 K TIT
Base Engine	890.0	890.0	890.0	890.0	890.0
Combustor and Fuel System	---	20.0	20.0	20.0	20.0
Regen. (2)	---	168.8	160.4	160.4	160.4
Recuperator	250.0	---	---	---	---
Intercoolers	20.0	20.0	20.0	20.0	20.0
Ducting	3.0	8.0	8.0	8.0	8.0
Turbo-machinery Mods	---	---	111.9	---	111.9
Total Initial Cost	1163	1106.8	1210.3	1098.4	1210.3

The turbomachinery-modification costs are added to the base engine cost. This assumes, therefore, that the base 5650 engine is purchased then modified. In addition, the costs listed are for production and do not include development costs. A new engine design requires many years and millions of dollars to develop.

On the other hand, the production cost of most items decreases rapidly with units made. The initial and replacement cost of the ceramic heat-exchanger cores seems particularly open to large price reductions, because they are manufactured generally by an extrusion process favoring automatic control A summary of the variable life-cycle-cost inputs is shown in Table 9.3.

Table 9.3 Life-Cycle Cost Variables

	kW	E	I (000 omitted)
Base 5650	2770	0.336	1140.0
IC Base 5650	3520	0.363	1163.0
IC EH Base 5650	3257	0.356	1106.8
IC EH Redesign	2961	0.408	1210.3
IC EH Off-Design	1875	0.409	1098.4
IC EH 1339 K TIT	3377	0.433	1210.3

The results of inserting the terms from Tables 9.1 and 9.3 into the life-cycle-cost equation (9.1) are summarized in Table 9.4.

Table 9.4 Life-Cycle Cost Summary ($/kWhr $\times 10^{2}$)

	Capital Cost	Fuel Cost	Maintenance	Life-Cycle Cost
Base 5650	0.113	3.575	0.500	3.997
IC Base 5650	0.385	2.773	0.500	3.678
ICEHBase5650	0.409	1.783	1.000	3.273
ICEH Redesign	0.492	1.556	1.000	3.236
ICEH Off-Des.	0.705	1.552	1.000	4.134
IC EH 1339 K	0.4311	1.466	1.000	2.940

Despite having the highest initial cost, the intercooled exhaust-heated engine possesses the lowest life-cycle cost of all the configurations considered (see figure 9.1). The turbomachinery redesign is the most cost-effective solution to running the intercooled exhaust-heated engine at the optimal pressure ratio. The off-design option has the highest life-cycle cost due to the low power output at the optimal pressure ratio. Running the intercooled exhaust-heated 5650 at its original design pressure ratio presents a favorable comparison to the redesign of turbomachinery due to its lower initial capital cost. For the exhaust-heated engines, increasing turbine inlet temperature produces great economic benefits over the life-cycle of the engines.

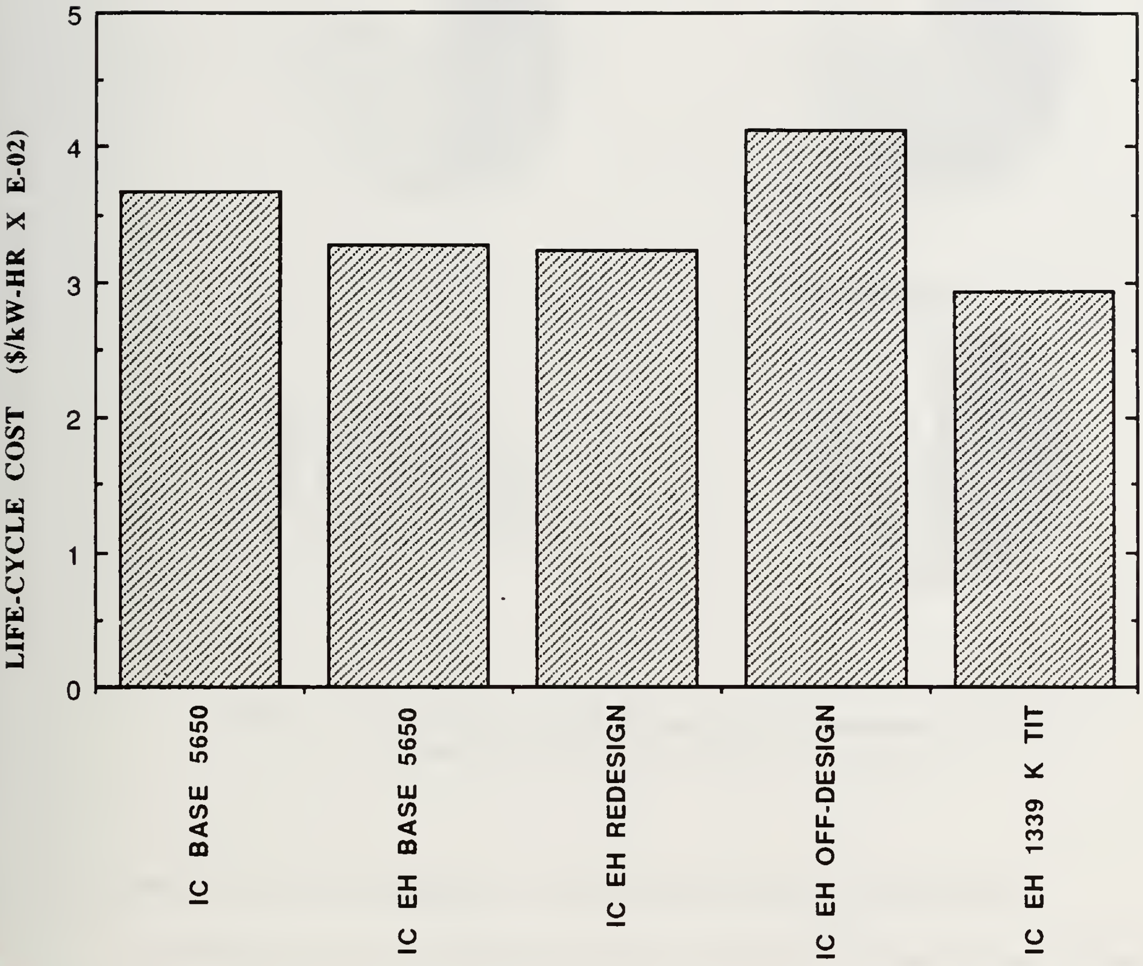


Figure 9.1 Life-Cycle Cost

The relative cost in $/kW-hr for the ten configurations examined is displayed in figure 9.2. Although the life-cycle fuel cost of the intercooled and non-intercooled base 5650 is substantially higher relative to all the alternatives, the overall life-cycle cost is low because initial capital and maintenance costs are small. The off-design options appear too expensive per kilowatt-hr. to purchase and maintain.

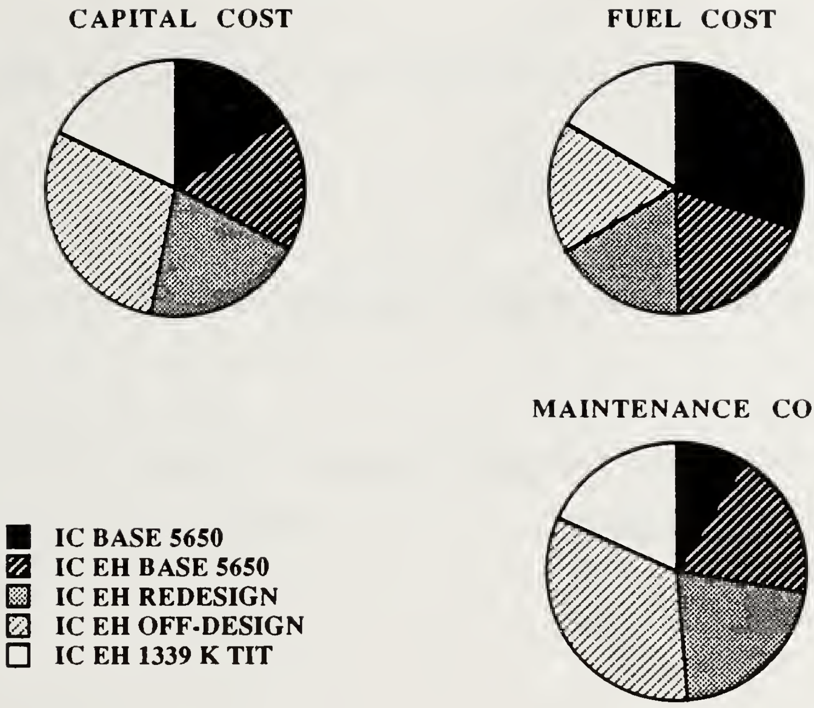


Figure 9.2 Relative Life-Cycle Cost Composite Charts (Intercooled Exhaust-Heated Cycles

9.3 Blue Sky And Optimal Off The Shelf Design Comparison

In a previous report, an optimal "blue Sky" intercooled exhaust-heated engine design was developed and evaluated by Tampe [40]. This design is designated CICXEB while the optimal conversion design of the intercooled-base-5650 is designated ICEH Redesign. Table 9.5 provides a normalized cost comparison of the two designs.

Table 9.5 Life-Cycle Cost Summary ($/kWhr x 10^{2})

	Capital Cost	Fuel Cost	Maintenance	Life-Cycle Cost
ICEH Redesign	0.492	1.556	1.000	3.236
CICXEB	0.648	1.244	1.000	2.893

Table 9.6 compares output power, thermal efficiency, and the initial capital cost of the respective engines.

Table 9.6 Comparison of Life-Cycle Cost Variables

	kW	E	I (000 omitted)
IC EH Redesign	2961	0.408	1210.3
CICXEB	2000	0.510	1078.0

Finally, Table 9.7 shows the respective component efficiencies for both of the optimized designs.

Table 9.7 CICXEB and ICEH Redesign Component Comparison

	CICXEB	ICEH Redesign
Compressor 1st Stage or Axial Comp. #1		
Efficiency (%)	91.8	84.1
Intercooler		
Effectiveness	.90	.902
Compressor 2nd Stage or Axial Comp. #2		
Efficiency (%)	91.8	79.6
Regenerator		
Effectiveness	0.975	0.975
Combustor		
Efficiency (%)	95	95
Gas-Producer Turbine		
Efficiency (%)	-------	88.7
Power Turbine		
Efficiency (%)	.913	87.7

Although normalized capital costs are less for the converted Solar 5650, the initial engine cost is actually greater due to its larger power output over the CICXEB design. The thermal efficiency comparison shows that the ICEH Redesign engine performs at a full 10 percentage points less than the optimal "blue sky" design. The reason for this is seen in the component efficiency comparisons where the differences in polytropic compressor efficiencies are quite substantial. The CICXEB design incorporated two, 3-stage, axial compressors while the performance of the ICEH Redesign is modelled after the tested design performance of the two stage centrifugal compressor found in the Solar 5650 engine. The substantial difference in life-cycle costs is a direct result of the higher

projected component efficiencies of the "blue sky" engine. The Probability of attaining these component efficiencies is much more difficult to assess when compared to the conversion of an existing engine with documented performance. Conversion and redesign of the 5650 engine appears to be a lower risk project because of existing components and and the possibility of a shorter time frame of project completion. This lower risk also manifests itself with a fairly large sacrifice in efficient performance. A final recommendation for which design option is most attractive depends greatly upon the level of risk , available capital, and project duration constraining the project manager.

10.0 COMBUSTOR RECOMMENDATIONS

As stressed earlier, the major advantage of the exhaust-heated cycle over conventional direct-fired units is that no products of combustion pass through the turbine. The rotary regenerator must tolerate the various emissions from the chosen combustor. Still, the economic viability of this cycle is dependent on using coal in its most inexpensive and untreated state. Many programs have been primarily investigating coal-water slurry (CWS) and beneficiated grades of coal. As fuel-treatment costs, and hot-gas-cleanup costs increase, prohibitively high life-cycle costs could eventually degrade the economic benefits of using coal over petroleum or natural gas. After researching the various types of combustors available for the exhaust-heated cycle, one developmental model stands out as a practical and effective component.

Avco Research Laboratory / Textron (ARL) has been developing and testing a slagging combustor for use in a direct coal-fired 80 MW gas turbine. It is unique in that standard utility-grade coal is fed into the primary zone of the combustor using pressurized air. All the testing has employed pulverized coal which is loaded into a conical bottom tank which is pressurized with dry nitrogen. The coal is fluidized with nitrogen introduced near the bottom of the cone. At the outlet of the tank is an orifice through which the coal flow is metered into a carrier line. Flow rate is then adjusted by altering the pressure difference between the tank and the carrier line at the orifice [45].

Pretreatment of the coal is practically eliminated and excessive pollutants and particulates are minimized. This is accomplished by burning the coal at a temperature higher than its ash melting point and removing the molten slag with an impact separator. The combustor has a primary rich-burn zone followed by a secondary lean-burn zone which produces very low NO_X emissions. A limestone sorbent is injected into the primary zone to control sulfur oxides.

Extensive testing of the combustor has led to the conclusion that additional cleanup stages may be necessary due to some of the particulate emissions which could potentially erode and foul the turbine in the direct-fired cycle. Currently, particulate and sulfur reduction has not been reduced to the level required to meet the EPA's New Source Performance Standards (NSPS) for coal-fired plants. Even if these levels are met, turbine erosion and fouling will still be an important issue. Alkalinity of the exhaust gases has not been fully investigated but preliminary analyses of slag samples indicate that approximately 80% of the alkali present in the coal is retained in the slag [46]. Obviously, some of the stringent constraints required by the direct-fired cycle above and beyond those necessary to meet pollution standards are reduced or even eliminated by the exhaust-heated cycle if an effective plan for cleaning or replacing rotary regenerators is implemented.

11.0 CONCLUSIONS AND RECOMMENDATIONS

This study has demonstrated the technical feasibility and benefits of converting an "off-the-shelf" gas turbine to an intercooled and non-intercooled exhaust-heated, coal-burning engine. Thorough cycle analysis produced the optimal pressure ratio for the converted engines and three options were presented to modify the intercooled and non-intercooled Solar "base 5650" engine to achieve the desired performance. Each of these options as well as an increased-turbine-inlet-temperature modification were examined on a life-cycle-cost basis. To achieve maximum benefit from the exhaust-heated cycle, both intercooling and increased-turbine-inlet-temperature modifications studied briefly here should be further scrutinized to determine their feasibility.

There are several areas which need more investigation before a decision could be made to modify a Solar 5650 engine. A demonstration of ceramic-rotary-regenerator performance under simulated coal-exhaust conditions is very important to the success of the engine. The effectiveness, seal leakage, wear, and ash-clogging tendency could all be quantified with a simple test rig. In addition, more research should be funded for atmospheric-slagging combustors and ash-clean-up systems. The current DOE emphasis is on direct-fired gas turbines where combustion occurs under high pressure. These high-pressure combustors must use coal injected in a coal-water slurry. Atmospheric combustors, on the other hand, can burn powdered coal and do not need an elaborate fuel-injection system. Finally, the environmental aspects of coal combustion must be further examined. Alternative fuels, such as biomass, which could be readily adapted to this cycle should also be investigated A recent presentation of studies done by Professor J. Beer concluded that a combination of clean-combustion technologies, energy-efficient power cycles and selective use of natural gas could provide environmentally safe energy [47]. The combustion technology to guarantee this, however, is still in its infancy.

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APPENDIX 1

Intercooled Solar 5650 Cycle
Computer Model

```
10  REM ************************************************************************
20  REM     THIS PROGRAM WILL CALCULATE THE EFFICIENCY AND SPECIFIC
30  REM     POWER FOR THE INTERCOOLED BASE SOLAR 5650.  THE PROGRAM ACCOUNTS
40  REM     FOR VARIATIONS IN POLYTROPIC EFFICIENCY OF THE VARIOUS
50  REM     COMPONENTS AS WELL AS THE MASS FLOW RATE LOSSES THROUGH THE
60  REM     HEAT EXCHANGER   WRITTEN BY LUIS TAMPE, ALTERED BY DAVID KOWALICK
65  REM     AND HARRY NAHATIS   LAST REVISION  4 10,90
70  REM ************************************************************************
74 LPRINT ; CHR$(15
80  KEY OFF
81  CLS
82  PRINT "PLEASE INPUT THE NAME OF THE DATA FILE "B:NAME.DAT "
83  INPUT B$
84  OPEN B$ FOR OUTPUT AS #1
85  CLS
86 PRINT "PLEASE INPUT THE DESIRED EFFECTIVENESS."
87 INPUT EFFEC
90 REM*************PHYSICAL CONSTANTS***************************
100 RG = .28696: PI = 3.141592654#
102 REM*************INLET CONDITIONS*****************************
106 PAT = 101.325
108 WA1 = 13.77
110 T1 = 288
127 REM********COOLING FLOWS AND PRESSURE LOSSES*****************
128 DP1 = .01: DP3 = .043: DP5 = .0157: DP6 = .06: DP7 = .0061: DP8 = .004
130 DELP = DP1 - DP3 - DP5 - DP6 - DP7 + DP8
132 DELPC = .0058
134 DELPH = .0629
136 WACL5 = .025 * WA1: WACL6 = 3.000001E-03 * WA1: WACL4 = .004 * WA1
139 REM*************BEGIN CALCULATION OF CYCLE PARAMETERS********
140  P2Q1 = 7.1
150 REM ****************** COMPRESSOR ***************************
155 P11Q1 = SQR(1.085 * P2Q1): PRSTAGE = P11Q1
157 E11Q1 = .8412:  EFFSTAGE = E11Q1
160 TEMP1 = T1
170 TEMP2 = 0
180 TAV = TEMP1
200 GOSUB 4000
205 CP1 = CP:  CPST1 = CP1
220 EXP1 = RG / (CP1 * E11Q1)
230 TEMP3 = TEMP1 * P11Q1 ^ EXP1
250 GOSUB 9000
270 T11 = TEMP2
    REM *******************INTERCOOLER*************************************
    EFFECI = .902
    T13 = 302.4: REM 85F
    T12 = T11 - EFFECI * (T11 - T13)
    REM*********************COMPRESSOR STAGE 2*****************************
    P2Q12 = P11Q1 / 1.085: PRSTAGE = P2Q12
    E2Q12 = .79632: EFFSTAGE = E2Q12
    TEMP1 = T12
    TEMP2 = 0
    TAV = TEMP1
    GOSUB 4000
```

```
    CP1 = CP:  CPST2 = CP1
    EXP1 = RG / (CP1 * E2Q12)
    TEMP2 = TEMP1 * P2Q12 ^ EXP1
    GOSUB 9000
    T2 = TEMP2
    TAV = .5 * (T1 + T2)
    GOSUB 4000
    CP1A2 = CP
    PWRCOMP = WA1 * (CPST1 * (T11 - T1) + CPST2 * (T2 - T12))
    WA2 = WA1: WA21 = WA1 - WACL5 - WACL6 - WACL4
    WA3 = WA21
340 REM **********GAS PRODUCER TURBINE*****************
350 E5Q4 = .8785
356 T4 = 1241.33
360 PWRCT = PWRCOMP: NN = 0
365 DT5 = 0: T5 = T4: TAV = T4: GOSUB 4000
366 WHILE ABS(DT5 - T5) > .5
367 NN = NN + 1
368 IF NN = 1 THEN GOTO 370
369 T5 = DT5
370 CP4A5 = CP: DT5 = T4 - PWRCT / (CP4A5 * WA21)
371 TAV = .5 * (T4 - DT5): GOSUB 4000
372 WEND
374 T5 = DT5: TDELP = DELP - DELPC - DELPH
375 P4Q6 = P2Q1 * (1 - TDELP)
376 P4Q5 = (T5 / T4) ^ (-CP4A5 / (RG * E5Q4))
377 REM*************POWER TURBINE**********************
378 P5Q6 = P4Q6 / P4Q5
379 TEMP3 = 0: TEMP2 = 0: E6Q5 = .8686
380 TAV = T5:  GOSUB 4000
381 CP5 = CP: EXP2 = -(RG * E6Q5) / CP5
    COOL = 1.01: REM COOLING AIR FACTOR
    T51 = T5 / COOL
382 TEMP1 = T51
383 TEMP2 = TEMP1 * P5Q6 ^ EXP2
384 GOSUB 9500
385 CP5A6 = CP: T6 = TEMP2
480 REM ****************** COMBUSTOR ***********************
481 E4Q3 = .95
490 FHV = 41600
495 T3 = T2 - EFFEC * (T6 - T2)
500 TAV = .5 * (T3 - T4
510 GOSUB 4000
515 CP3A4 = CP
520 QIN = (CP3A4 * (T4 - T3) * WA3) / E4Q3
530 WF = QIN / FHV
532 TAV = .5 * (T6 - T3): GOSUB 4000
534 CP2A6 = CP
535 W4 = WA3 + WF - WACL4: W5 = W4 - WACL5: W6 = W5 - WACL6: W7 = W6
537 REM *************PERFORMANCE AND MISC. CALCS. ****************
540 PWRPT = W5 * CP5A6 * (T5 - T6)
545 PWRNET = PWRPT * .985
550 ETATH = PWRNET / QIN
560 PWRSPEC = PWRNET / (CP1 * T1 * WA1)
```

```
565 SFC = (WF * 3600) / PWRNET
570 FARAT = WF / WA3
580 REM ****************** RECUPERATOR ****************************
585 TAV = .5 * (T3 + T2): GOSUB 4000
590 CP2A3 = CP
595 CC = WA2 * CP2A3
600 QC = CC * (T3 - T2)
602 T7 = T6: DT7 = 0
604 WHILE ABS(DT7 - T7) > .5
605 DT7 = T7
609 TAV = .5 * (T7 + T6): GOSUB 4000
610 CP6A7 = CP
615 CH = W6 * CP6A7
620 T7 = T6 - (QC / CH)
625 WEND
630 REM **************** OUTPUT ********************************
660 LPRINT , "INTERCOOLED BASE SOLAR 5650 PROGRAM SOLAR5A.BAS": LPRINT , " "
662 LPRINT , "OUTPUT FILE ="; B$: LPRINT , " "
664 LPRINT , "INPUT SUMMARY": LPRINT , " "
668 LPRINT , "RECUPERATOR EFFECTIVENESS ="; EFFEC
670 LPRINT , " ": LPRINT , "PERFORMANCE SUMMARY"
671 LPRINT , " ": LPRINT , "NET POWER OUTPUT (KW) ="; PWRNET
672 LPRINT , "THERMAL EFFICIENCY ="; ETATH
673 LPRINT , "SPECIFIC FUEL CONSUMPTION (KG/KW/HR) ="; SFC
674 LPRINT , "SPECIFIC POWER (KW/KG) ="; PWRSPEC
675 LPRINT , " ": LPRINT , "COMPONENT SUMMARY": LPRINT , " ": LPRINT , "COMPRESSOR"
    LPRINT , " ": LPRINT , "FIRST STAGE"
676  LPRINT , "STATION          1        11"
677 LPRINT , "MASSFLOW (KG/SEC)"; WA1, WA1
678 LPRINT , "TEMPERATURE (DEG K)"; T1; T11
679 LPRINT , "EFFICIENCY ="; E11Q1
680 LPRINT , "PRESSURE RATIO ="; P11Q1
    LPRINT , " ": LPRINT , "INTERCOOLER"
    LPRINT , "EFFECTIVENESS ="; EFFECI
    LPRINT , "STATION          11       12 "
    LPRINT , "TEMPERATURE (DEG K)"; T11; T12
    LPRINT , " ": LPRINT , "SECOND STAGE"
    LPRINT , "STATION          12       2"
    LPRINT , "MASSFLOW (KG/SEC)"; WA1, WA1
    LPRINT , "TEMPERATURE (DEG K)"; T12; T2
    LPRINT , "EFFICIENCY ="; E2Q12
    LPRINT , "PRESSURE RATIO =", P2Q12
681 LPRINT , "POWER (KW) =", PWRCOMP
    LPRINT , "COMPRESSOR PRESSURE RATIO", P2Q1
682 LPRINT , " ": LPRINT , "GAS PRODUCER TURBINE"
683 LPRINT , " ": LPRINT , "STATION           4        5"
684 LPRINT , "MASSFLOW (KG/SEC)", W4, W5
690 LPRINT , "TEMPERATURE (DEG K)", T4; T5
695 LPRINT , "EFFICIENCY", E5Q4
700 LPRINT , "PRESSURE RATIO =", P4Q5
705 LPRINT , "POWER (KW) =", PWRGT
710 LPRINT , "PRESSURE LOSS =", [illegible]
715 LPRINT , " ": LPRINT , "POWER TURBINE"
720 LPRINT , " ": LPRINT , "STATION           5        6"
```

```
725 LPRINT , "MASSFLOW (KG/SEC)", W5; W6
730 LPRINT , "TEMPERATURE (DEG K)"; T51; T6
735 LPRINT , "EFFICIENCY", E6Q5
740 LPRINT , "PRESSURE RATIO ="; P5Q6
745 LPRINT , "POWER (KW) =", PWRPT
750 LPRINT , "PRESSURE LOSS ="; DP6
755 LPRINT , " ": LPRINT , "COMBUSTOR"
756 LPRINT , " ": LPRINT , "STATION          3        4"
757 LPRINT , "MASSFLOW (KG/SEC)"; WA3; W4
758 LPRINT , "TEMPERATURE (DEG K)"; T3; T4
760 LPRINT , "EFFICIENCY ="; E4Q3
765 LPRINT , "PRESSURE LOSS ="; DP3
770 LPRINT , "FUEL FLOW (KG/HR) ="; WF * 3600
775 LPRINT , "FUEL-AIR RATIO ="; FARAT
777 LPRINT , "FUEL HEATING VALUE (KJ/KG) ="; FHV
780 LPRINT , " ": LPRINT , "RECUPERATOR"
785 LPRINT , " ": LPRINT , "EFFECTIVENESS =", EFFEC
830 LPRINT , " ": LPRINT , "COLD SIDE"
835 LPRINT , "INLET TEMPERATURE (DEG K) =", T2
840 LPRINT , "EXIT TEMPERATURE (DEG K) ="; T3
850 LPRINT , "DELPC =", DELPC
851 LPRINT , " ": LPRINT , "HOT SIDE"
852 LPRINT , "INLET TEMPERATURE (DEG K) ="; T6
853 LPRINT , "EXIT TEMPERATURE (DEG K) =", T7
855 LPRINT , "DELPH =", DELPH
860 LPRINT , " ": LPRINT , "COOLING FLOWS AND LOSSES"
862 LPRINT , "COMBUSTOR COOLING (%WA1) ="; WACL4 / WA1 * 100
865 LPRINT , "GAS PRODUCER TURBINE COOLING (%WA1) ="; WACL5 / WA1 * 100
870 LPRINT , "POWER TURBINE COOLING (%WA1) =", WACL6 / WA1 * 100
872 LPRINT , "INLET/EXHAUST PRESSURE LOSSES =" DP1
875 LPRINT , "TOTAL PRESSURE LOSSES =", TDELP
876 CLS
910 END
4000 REM ********************************************************************
4010 REM    THIS SUBROUTINE WILL CALCULATE THE DENSITY, VISCOSITY,
4020 REM    AND PRANDTL NUMBER FOR AIR.  IT IS ACCURATE FOR A TEMPERATURE
4030 REM    RANGE BETWEEN 100K AND 1100K.
4040 REM ********************************************************************
4050 Z = TAT
4060 IF Z < 300 THEN GOTO 4470
4061 IF Z < 550 THEN GOTO 4200
4070 IF Z < 800 THEN GOTO 4300
4080 IF Z < 1100 THEN GOTO 4400
4081 IF Z < 1500 THEN GOTO 4000
4081 A1 = -400.1222D#: A2 = 1690.0155#: A3 = .6430406#
4082 B1 = 1.0550014#: B2 = -5.8700112#: B3 = .0000026877#
4084 C1 = -.0000070195#: C2 = 5.455014700000001D-10: C3 = -.00000000142677#
4085 D1 = .0000000048002#: D2 = -6.700000E-08: D3 = 0
4086 E1 = -6.5815400E-11: E2 = .0000000012085557#: E3 = 0
4087 F1 = 1.5000800D-13: F2 = -6.4081000D-16: F3 = 0
4088 G1 = -1.4300000D-17: G2 = 5.0970000E-19: G3 = 0
4089 GOTO 4500
4090 REM *********************************************************
4100 A1 = -6.0240000999999999#: A2 = -111.5712#: A3 = -01.540504#
```

```
4100 B1 = .035373512#: B2 = .54300117#: B3 = .12123425#
4110 C1 = -.000054487517#: C2 = -.00081025676#: C3 = -.00017604929#
4120 D1 = .000000042125013#: D2 = .00000060500505#: D3 = .00000012729322#
4130 E1 = -1.6250005D-11: E2 = -2.2500188D-10: E3 = -4.5838964D-11
4140 F1 = 2.5000007D-15: F2 = 3.3333612D-14: F3 = 6.5780793D-15
4150 G1 = 0!: G2 = 0!: G3 = 0!
4160 GOTO 4500
4170 REM ********************************************************
4200 A1 = 6.2422#: A2 = 125.98218#: A3 = 2.3111608#
4210 B1 = -7.5950233000000001D-02: B2 = -1.6405946#: B3 = -.019682303#
4220 C1 = .00045406911#: C2 = 8.9202883000000001D-03: C3 = .00010031399#
4230 D1 = -.00000143365#: D2 = -.000025608046#: D3 = -.00000026846625#
4240 E1 = .0000000025221111#: E2 = .000000041060078#: E3 = 3.9002344D-10
4250 F1 = -2.3426666D-12: F2 = -3.488007E-11:    F3 = -2.8672807D-13
4260 G1 = 8.9777777D-16: G2 = 1.2266693D-14: G3 = 8.1507873D-17
4270 GOTO 4500
4280 REM ********************************************************
4300 A1 = -377.3956#: A2 = -165.27518#: A3 = -6.0256168#
4310 B1 = 3.2763116#: B2 = 1.4255922#. B3 = .051037652#
4320 C1 = -.011771945#: C2 = -.005050252#: C3 = -.00028296231#
4330 D1 = .000022468439#: D2 = 9.5268121000000001D-06: D3 = 8.3510491000000001D-07
4340 E1 = -.000000024025964#: E2 = -.000000010074254#: E3 = -.000000001004582#
4350 F1 = 1.3646107D-11: F2 = 5.6574076D-12: F3 = 6.0120400D-13
4360 G1 = -3.0178002D-15: G2 = -1.0184349D-15: G3 = -1.6022768D-16
4370 GOTO 4500
4380 REM ********************************************************
4400 A1 = -71.104772999999999#: A2 = 1393.6403#: A3 = 0.4271042#
4410 B1 = .4642101 5#: B2 = -8.5703026#: B3 = -.011591627#
4420 C1 = -.0012087529#. C2 = .02167150 8#. C3 = .000013080706#
4430 D1 = .0000017529667#: D2 = 0 - .000029586195#: D3 = -.000000001203421#
4440 E1 = -.0000000013886662#: E2 = .000000022385540#. E3 = 0.1160572D-12
4450 F1 = 5.6281468D-13: F2 = -6.980678E-12: F3 = 0!
4460 G1 = -1.0129515D-16: G2 = 1.4526305E-15. G3 = 0!
4465 GOTO 4500
4470 REM ********************************************************
4471 A1 = 1.2066: A2 = 16.0569: A3 = 1.003
4472 B1 = -.0006769: B2 = -.36679893#  B3 = -.005682
4473 C1 = .000026998333#. C2 = 4.22705E-03: C3 = .0000502
4474 D1 = -.000000096500033#: D2 = -.000021464000#. D3 = -1.5E-07
4475 E1 = 1.7666667D-10: E2 = 5.1888E-08. E3 = 1.6E-10
4476 F1 = -1.1266667D-13: F2 = -4.7997000D-11  F3 = -4.8E-13
4477 G1 = 0. G2 = 0  G3 = 0
4478 REM ********************************************************
4500 X2 = X ^ 2
4510 X3 = X ^ 3
4520 X4 = X ^ 4
4530 X5 = X ^ 5
4540 X6 = X ^ 6
4550 CP = A1 + B1 * X + C1 * X2 + D1 * X3 + E1 * X4 + F1 * X5 + G1 * X6
4560 CP = CP * 10000
4570 CP = INT(CP + .5)
4580 CP = CP / 10000
4590 VIS = A2 + B2 * X + C2 * X2 + D2 * X3 + E2 * X4 + F2 * X5 + G2 * X6
4600 VIS = VIS * 1000
```

```
4610 VIS = INT(VIS + .5)
4620 VIS = VIS / 1E+08
4630 PR = A3 + B3 * X + C3 * X2 + D3 * X3 + E3 * X4 + F3 * X5 + G3 * X6
4640 PR = PR * 1000
4650 PR = INT(PR + .5)
4660 PR = PR / 1000
4670 RETURN
9000 REM **********************************************************************
9010 REM     THIS SUBROUTINE WILL ITERATE UNTIL THE TWO COMPRESSOR
9020 REM     TEMPERATURES ARE ONLY 0.5 DEGREES APART
9030 REM **********************************************************************
9040 NN = 0
9050 WHILE ABS(TEMP2 - TEMP3) > .5
9060 NN = NN + 1
9070 IF NN = 1 THEN GOTO 9090
9080 TEMP3 = TEMP2
9090 TAV = .5 * (TEMP1 + TEMP3)
9100 GOSUB 4000
9110 EXP1 = RG / (CP * EFFSTAGE)
9120 TEMP2 = TEMP1 * PRSTAGE ^ EXP1
9130 WEND
9140 RETURN
9500 REM **********************************************************************
9510 REM     THIS SUBROUTINE WILL ITERATE UNTIL THE TWO EXPANDER
9520 REM     TEMPERATURES ARE ONLY 0.5 DEGREES APART
9530 REM **********************************************************************
9540 NN = 0
9550 WHILE ABS(TEMP2 - TEMP3) > .5
9560 NN = NN + 1
9570 IF NN = 1 THEN GOTO 9590
9580 TEMP3 = TEMP2
9590 TAV = .5 * (TEMP1 + TEMP3)
9600 GOSUB 4000
9610 EXP1 = -(RG * E6Q5) / CP
9620 TEMP2 = TEMP1 * P5Q6 ^ EXP1
9630 WEND
9640 RETURN
```

APPENDIX 2

Intercooled Exhaust-Heated Cycle
Computer Model

```
10  REM ************************************************************************
20  REM     THIS PROGRAM WILL CALCULATE THE EFFICIENCY AND SPECIFIC
30  REM     POWER FOR THE INTERCOOLED SOLAR 5650, MODIFIED FOR EXHAUST HEATING.
40  REM     THE PROGRAM ACCOUNTS FOR VARIATIONS IN POLYTROPIC EFFICIENCY OF THE VARIOUS
50  REM     COMPONENTS AS WELL AS THE MASS FLOW RATE LOSSES THROUGH THE
60  REM     HEAT EXCHANGER.  WRITTEN BY LUIS TAMPE. ALTERED BY DAVID KOWALICK
65  REM     AND HARRY NAHATIS   LAST REVISION: 4/10/90
70  REM ************************************************************************
75  LPRINT ; CHR$(15)
80  KEY OFF
81  CLS
82  PRINT "PLEASE INPUT THE NAME OF THE DATA FILE (F:NAME.DAT)", F$
83  INPUT F$
85  CLS
87  PRINT "PLEASE INPUT THE DESIRED CORE TYPE :"
88  PRINT "1 - CORE 503A"
89  PRINT "2 - CORE 504A"
90  PRINT "3 - CORE 505A"
91  INPUT CORE
92 PRINT "PLEASE INPUT THE NUMBER OF REGENERATOR DISCS DESIRED:"
93 INPUT ND
95 MFRAC1 = .9375
96  PRINT "INPUT INITIAL CRAT GUESS (.950<CRAT<0.998)"
97  INPUT CRAT
98 PRINT "PLEASE INPUT THE DESIRED EFFECTIVENESS:"
99 PRINT "0.90, 0.95, OR 0.975."
100 INPUT EFFEC
102 REM **************** PHYSICAL CONSTANTS ***************************
104 RG = .28696: PI = 3.141592654#
105 REM *************INLET CONDITIONS ********************************
106 PAT = 101.325
108 WA1 = 16.77
110 T1 = 288
111 REM *************COOLING FLOWS AND PRESSURE LOSSES *****************
128 DP1 = 0!: DP2 = .040: DP5 = .0197: DP6 = .06: DP7 = .0361: DP8 = .004: DPDUCT = .04
130 DELP = DP1 - DP2 - DP5 - DP6 - DP7 + DP8 - DPDUCT
132 WACL21 = .025 * WA1: WACL4 = 8.000001E-03 * WA1: WACL5 = .004 * WA1
135 REM ************ BEGIN CALCULATION OF CYCLE PARAMETERS **************
140  P2Q1 = 4.5
145  PASO = .001: SIGNO = 1: REM PARAMETERS FOR CRAT ITERATION
150 REM ****************** COMPRESSOR - STAGE 1 *********************************
155 P11Q1 = SQR(1.065 * P2Q1): PRSTAGE = P11Q1
157 E11Q1 = .9011 - (P2Q1 - 1) / 100: EFFSTAGE = E11Q1
160 TEMP1 = T1
170 TEMP2 = 1
180 TAV = TEMP1
200 GOSUB 4000
205 CP1 = CP: CPST1 = CP1
210 EXP1 = RG / (CP1 * E11Q1)
220 TEMP2 = TEMP1 * P11Q1 ^ EXP1
250 GOSUB 9000
270 T11 = TEMP2
    REM ******************INTERCOOLER***********************************
    EFFEC1 = .900
```

```
T13 = 302.4: REM 85F
T12 = T11 - EFFECI * (T11 - T13)
REM********************COMPRESSOR STAGE 2*****************************
P2Q12 = P11Q1 / 1.065: PRSTAGE = P2Q12
E2Q12 = .85732 - ((P2Q1 - 1) / 100): EFFSTAGE = E2Q12
TEMP1 = T12
TEMP2 = 0
TAV = TEMP1
GOSUB 4000
CP1 = CP: CPST2 = CP1
EXP1 = RG / (CP1 * E2Q12)
TEMP3 = TEMP1 * P2Q12 ^ EXP1
GOSUB 9000
T2 = TEMP2
TAV = .5 * (T1 + T2)
GOSUB 4000
CP1A2 = CP
PWRCOMP = WA1 * (CPST1 * (T11 - T1) - CPST2 * (T2 - T12))
280 REM ********************REGENERATOR************************************
282 T5 = 1265
290 EMAX = EFFEC * (T5 - T2)
320 IF CRAT < 1 THEN GOTO 323
321 T6 = T5 - EMAX
322 T3 = T2 - (T5 - T6) / CRAT: GOTO 325
323 T3 = T2 - EMAX
324 T6 = T5 - CRAT * (T3 - T2)
325 WA21 = WA1 - WAC131 - WACL4
328 WA21QND = WA21 / ND
330 GOSUB 6000
331 OMEGA = (1 / ANGSPED) * 2 * PI
333 TQ = -.7467567#3 * DIAM + 2.7914622# * DIAM ^ 2: REM IN KNm
334 PWRREG = TQ * OMEGA * ND
335 PWRLOSS = 2.746 + PWRREG
336 GOSUB 7000
337 WA3 = WA21 - (WALT - WACT)
338 IF ABS(MFRAC2 - MFRAC1) < .0001 GOTO 350
339 MFRAC1 = MFRAC2
340 GOTO 325
349 REM ************* GAS PRODUCER TURBINE ******************************
350 E31Q3 = .8988 - ((P2Q1 - 1) / 300)
360 PWRCT = PWRCOMP: NN = 1
365 DT31 = 0: T31 = T3: TAV = T3: GOSUB 4000
366 WHILE ABS(DT31 - T31) > .5
367 NN = NN + 1
368 IF NN = 1 THEN GOTO 370
369 T31 = DT31
370 CP3A31 = CP: DT31 = T3 - PWRCT / (CP3A31 * WA3)
371 TAV = .5 * (T3 + DT31): GOSUB 4000
372 WEND
373 T31 = DT31: TDELP = DELP - (DELPC - DELPH) / 100
374 P3Q4 = P2Q1 * (1 - TDELP)
375 P3Q31 = (T31 / T3) ^ (-CP3A31 / (RG * E31Q3))
376 REM ****************POWER TURBINE**********************************
377 P31Q4 = P3Q4 / P3Q31: E4Q31 = .8989 - ((P2Q1 - 1) / 300)
```

```
378 TEMP3 = 0
379 TEMP2 = 0
380 TAV = T31: GOSUB 4000
381 CP31 = CP: EXP2 = -(RG * E4Q31) / CP31
    T31A = T31 / 1.01
382 TEMP1 = T31A
383 TEMP3 = TEMP1 * P31Q4 ^ EXP2
384 GOSUB 9500
385 T4 = TEMP2
386 TAV = .5 * (T31 + T4)
387 WA31 = WA3 + WACL31: WA4 = WA31 - WACL4: GOSUB 4000
388 CP31A4 = CP: PWRPT = (T31 - T4) * CP31A4 * WA31
389 PWREXP = PWRPT
490 REM ****************** COMBUSTOR *********************************
496 FHV = 342621
498 E5Q4 = .95
499 COALRAT = .96: REM COAL PORTION THAT IS USED AS GAS
500 TAV = .5 * (T4 + T5)
510 GOSUB 4000
515 CP4A5 = CP
520 W5 = W5QND * ND
525 WREGAVG = .5 * (W5 + WA31)
526 QIN = (CP4A5 * (T5 - T4) * WREGAVG) / E5Q4
530 WF = QIN / FHV
535 REM ******************PREFORMANCE AND MISC. CALCS.******************
540 PWRNET = (PWREXP - PWRLOSS) * .985
550 ETATH = PWRNET / QIN
560 PWRSPEC = PWRNET / (CP1 * T1 * WA1)
565 SFC = (WF * 3600) / PWRNET
566 ER = W5 - (WA4 - COALRAT * WF): FARAT = (W5 - WA4) / WA4
568 PRINT "SUCCESSFUL PASS,ER,CRAT "; ER; CRAT
569 IF ABS(ER) < .002 THEN GOTO 590
570 IF ER > 0 GOTO 580
571 IF SIGNO = 0 THEN PASO = PASO / 2
572 CRAT = CRAT - PASO
573 SIGNO = 1
575 GOTO 310
580 IF SIGNO = 1 THEN PASO = PASO / 2
582 SIGNO = 0
583 CRAT = CRAT - PASO
585 GOTO 310
588 REM *****************POWER ITERATION (NOT USED)*********************
589 REM PRINT " ": PRINT "ADJUSTING MASS FLOW TO PRODUCE REQUIRED POWER"
591 GOTO 650
592 WA1CHK = 1000   PWRSPEC * CP1 * T1
595 IF ABS(WA1 - WA1CHK) < .0001 THEN GOTO 650
597 PWRCOMP = PWRCOMP * (WA1CHK / WA1)
600 SIGNO = 2
605 WA1 = WA1CHK
610 GOTO 310
650 REM ********************* OUTPUT *********************************
651 LPRINT , "EXHAUST-HEATED INTERCOOLED FASC PROGRAM INTCOOL.BAS"
652 LPRINT , "OUTPUT FILE =" F$: LPRINT , " "
653 LPRINT , "INPUT SUMMARY": LPRINT , " "
```

```
666 LPRINT , "REGENERATOR CORE TYPE ="; CORE
667 LPRINT , "NUMBER OF REGENERATOR DISCS ="; ND
668 LPRINT , "REGENERATOR EFFECTIVENESS ="; EFFEC
670 LPRINT , " ": LPRINT , "PERFORMANCE SUMMARY"
671 LPRINT , " ": LPRINT , "NET POWER OUTPUT (KW) ="; PWRNET
672 LPRINT , "THERMAL EFFICIENCY ="; ETATH
673 LPRINT , "SPECIFIC FUEL COMSUMPTION (KG/KW/HR) ="; SFC
674 LPRINT , "SPECIFIC POWER (KW/KG) ="; PWRSPEC
675 LPRINT , " ": LPRINT , "COMPONENT SUMMARY": LPRINT , " ": LPRINT , "COMPRESSOR"
    LPRINT , " ": LPRINT , "FIRST STAGE"
676  LPRINT , "STATION          1        11"
677 LPRINT , "MASSFLOW (KG/SEC)"; WA1; WA1
678 LPRINT , "TEMPERATURE (DEG K)"; T1; T11
679 LPRINT , "EFFICIENCY ="; E11Q1
680 LPRINT , "PRESSURE RATIO ="; P11Q1
    LPRINT , " ": LPRINT , "INTERCOOLER"
    LPRINT , "EFFECTIVENESS ="; EFFECI
    LPRINT , "STATION          11        12 "
    LPRINT , "TEMPERATURE (DEG K)"; T11; T12
    LPRINT , " ": LPRINT , "SECOND STAGE"
    LPRINT , "STATION          12        2"
    LPRINT , "MASSFLOW (KG/SEC)"; WA1; WA1
    LPRINT , "TEMPERATURE (DEG K)"; T12, T2
    LPRINT , "EFFICIENCY ="; E2Q12
    LPRINT , "PRESSURE RATIO ="; P2Q12
681 LPRINT , "POWER (KW) ="; PWRCOMP
    LPRINT , "COMPRESSOR PRESSURE RATIO"; P2Q1
682 LPRINT , " ": LPRINT , "GAS PRODUCER TURBINE"
683 LPRINT , " ": LPRINT , "STATION          3        31"
684 LPRINT , "MASSFLOW (KG/SEC)"; WA3, WA31
690 LPRINT , "TEMPERATURE (DEG K)"; T3, T31
695 LPRINT , "EFFICIENCY"; E31Q3
700 LPRINT , "PRESSURE RATIO ="; P3Q31
705 LPRINT , "POWER (KW) ="; PWRGT
710 LPRINT , "PRESSURE LOSS ="; DP5
715 LPRINT , " ": LPRINT , "POWER TURBINE"
720 LPRINT , " ": LPRINT , "STATION          31A        4"
725 LPRINT , "MASSFLOW (KG/SEC)"; WA31, WA4
730 LPRINT , "TEMPERATURE (DEG K)"; T31A, T4
735 LPRINT , "EFFICIENCY"; E4Q31
740 LPRINT , "PRESSURE RATIO ="; P31Q4
745 LPRINT , "POWER (KW) ="; PWRPT
750 LPRINT , "PRESSURE LOSS =", DP6
755 LPRINT , " ": LPRINT , "COMBUSTOR"
756 LPRINT , " ": LPRINT , "STATION          4        5"
757 LPRINT , "MASSFLOW (KG/SEC)", WA4, W5
758 LPRINT , "TEMPERATURE (DEG K)", T4, T5
760 LPRINT , "EFFICIENCY ="; E5Q4
765 LPRINT , "PRESSURE LOSS ="; DP3
770 LPRINT , "FUEL FLOW (KG/HR) =", WF * 3600
775 LPRINT , "FUEL-AIR RATIO ="; FARAT
777 LPRINT , "FUEL HEATING VALUE (KJ/KG) =", FHV
780 LPRINT , " ": LPRINT , "REGENERATOR"
785 LPRINT , " ": LPRINT , "EFFECTIVENESS ="; EFFEC
```

```
790 LPRINT , "POWER REQUIRED (KW) ="; PWRREG
800 LPRINT , "NUMBER OF DISKS ="; ND
805 LPRINT , "DIAMETER OF EACH DISK (M) ="; DIAM
807 LPRINT , "THICKNESS OF EACH DISK (M) ="; THK
810 LPRINT , "MASS OF EACH DISK (KG) ="; MASSMAT
815 LPRINT , "ANGULAR SPEED (RPM) ="; 1 / ANGSPED * 60
820 LPRINT , "TOTAL RADIAL SEAL LEAKAGE AND % ="; WALT; (WALT / WA21) * 100
825 LPRINT , "TOTAL CIRCUMF. LOSS AND % ON COLD SIDE ="; WACT; (WACT / WA21) * 100
830 LPRINT , " ": LPRINT , "COLD SIDE"
835 LPRINT , "INLET TEMPERATURE (DEG K) ="; T2
840 LPRINT , "EXIT TEMPERATURE (DEG K) ="; T3
850 LPRINT , "DELPC,AHC,AFFC,AFC ="; DELPC; AHC; AFFC; AFC
851 LPRINT , " ": LPRINT , "HOT SIDE"
855 LPRINT , "DELPH,AHH,AFFH,AFH ="; DELPH; AHH; AFFH; AFH
860 LPRINT , " ": LPRINT , "COOLING FLOWS AND LOSSES"
865 LPRINT , "GAS PRODUCER TURBINE COOLING (%WA1) ="; WACL31 / WA1 * 100
870 LPRINT , "POWER TURBINE COOLING (%WA1) ="; WACL4 / WA1 * 100
872 LPRINT , "INLET/EXHAUST LOSSES ="; DP1
873 LPRINT , "REGENERATOR DUCT LOSSES ="; DPDUCT
875 LPRINT , "TOTAL PRESSURE LOSSES ="; TDELP
876 CLS
880 PRINT "CALCULATIONS COMPLETE, OUTPUT ON FILE   "; B$
881 PRINT : PRINT "NET POWER OUTPUT (KW) ="; PWRNET
882 PRINT "THERMAL EFFICIENCY ="; ETATH
883 PRINT "SPECIFIC FUEL COMSUMPTION (KG/KW/HR) ="; SFC
884 PRINT "SPECIFIC POWER (KW/KG) ="; PWRSPEC
910 STOP
4000 REM **************************************************************************
4010 REM    THIS SUBROUTINE WILL CALCULATE THE DENSITY, VISCOSITY,
4020 REM    AND PRANDTL NUMBER FOR AIR.  IT IS ACCURATE FOR A TEMPERATURE
4030 REM    RANGE BETWEEN 100K AND 2100K.
4040 REM **************************************************************************
4050 X = TAV
4055 IF X < 300 THEN GOTO 4401
4060 IF X < 550 THEN GOTO 4200
4070 IF X < 800 THEN GOTO 4300
4080 IF X < 1100 THEN GOTO 4400
4081 IF X < 1500 THEN GOTO 4095
4082 A1 = -402.12209#: A2 = 1690.6066#: A3 = .64914286#
4083 B1 = 1.3956504#: B2 = -5.8736182#. B3 = .000069285714#
4084 C1 = -.0020070195#: C2 = 8.4892147000000001D-03: C3 = -.000000021428571#
4085 D1 = .0000015343312#: D2 = -6.518326E-06: D3 = 0
4086 E1 = -6.5815401D-10: E2 = .000000002805566 7#: E3 = 0
4087 F1 = 1.5002802D-13: F2 = -6.4182000D-13: F3 = 0
4088 G1 = -1.4200620D-17: G2 = 6.097068E-18: G3 = 0
4089 GOTO 4500
4090 REM ********************************************************
4095 A1 = -6.1240030999999999#: A2 = -141.9712#: A3 = -01.542504#
4100 B1 = .0050703512#: B2 = .54300117#: B3 = .1010040 5#
4110 C1 = -.0000544875 17#: C2 = -.0006102567 6#: C3 = -.000176049 29#
4120 D1 = .000000042115013#: D2 = .00000069150 503#: D3 = .0000001179011#
4130 E1 = -1.6250005D-11: E2 = -2.2800188D-10: E3 = -4.5803964D-11
4140 F1 = 1.5000007D-15: F2 = 3.3000612D-14. F3 = 6.5760790D-15
4150 G1 = 0!: G2 = 0!: G3 = 0!
```

```
4160 GOTO 4500
4170 REM ******************************************************
4200 A1 = 6.2422#: A2 = 125.98218#: A3 = 2.3111608#
4210 B1 = -7.595023300000001D-02: B2 = -1.6405946#: B3 = -.019682303#
4220 C1 = .00045406911#: C2 = 8.920288300000001D-03: C3 = .00010031399#
4230 D1 = -.00000143365#: D2 = -.000025608046#: D3 = -.00000026846625#
4240 E1 = .0000000025221111#: E2 = .000000041060078#: E3 = 3.9002344D-10
4250 F1 = -2.3426666D-12: F2 = -3.488007E-11:    F3 = -1.8672807D-13
4260 G1 = 8.9777777D-16: G2 = 1.2266693D-14: G3 = 8.1507873D-17
4270 GOTO 4500
4280 REM ******************************************************
4300 A1 = -377.3966#: A2 = -165.27528#: A3 = -8.0256168#
4310 B1 = 3.2763116#: B2 = 1.4255922#: B3 = .091037651#
4320 C1 = -.011771945#: C2 = -.0050502522#: C3 = -.00038296231#
4330 D1 = .000022468439#: D2 = 9.528912100000001D-06: D3 = 8.361049100000001D-07
4340 E1 = -.000000024025964#: E2 = -.000000010074254#: E3 = -.0000000010045823#
4350 F1 = 1.3648107D-11: F2 = 5.6574076D-12: F3 = 6.0220433D-13
4360 G1 = -3.2176032D-15: G2 = -1.3164349D-15: G3 = -1.6322768D-16
4370 GOTO 4500
4380 REM ******************************************************
4400 A1 = -71.10477299999999#: A2 = 1393.6423#: A3 = 3.4271042#
4410 B1 = .46421015#: B2 = -8.5703026#: B3 = -.011591627#
4420 C1 = -.0011087839#: C2 = .021671508#: C3 = .000018050706#
4430 D1 = .0000017509667#: D2 = 0 - .000023586195#: D3 = -.0000000012330421#
4440 E1 = -.0000000010886662#: E2 = .0000000022365546#: E3 = 3.1160570D-12
4450 F1 = 5.8261468D-13: F2 = -8.980678E-12: F3 = 0!
4460 G1 = -1.0129505D-16: G2 = 1.4926305D-15: G3 = 0!
4465 GOTO 4500
4470 REM ******************************************************
4471 A1 = 1.2064: A2 = 14.0569: A3 = 1.005
4472 B1 = -.0006769: B2 = -.36679890#: B3 = -.005681
4473 C1 = .000126996030#: C2 = 4.20706E-03: C3 = .0000531
4474 D1 = -.000000098500333#: D2 = -.000021464033#: D3 = -1.5E-07
4475 E1 = 1.7666667D-10: E2 = 5.1888E-08: E3 = 5.6E-11
4476 F1 = -1.2266667D-13: F2 = -4.7997333D-11: F3 = -4.8E-13
4477 G1 = 0: G2 = 0: G3 = 0
4478 REM ******************************************************
4500 X2 = X ^ 2
4510 X3 = X ^ 3
4520 X4 = X ^ 4
4530 X5 = X ^ 5
4540 X6 = X ^ 6
4550 CP = A1 - B1 * X - C1 * X2 - D1 * X3 - E1 * X4 - F1 * X5 - G1 * X6
4560 CP = CP * 10000
4570 CP = INT CP - .5
4580 CP = CP / 10000
4590 VIS = A2 - B2 * X - C2 * X2 - D2 * X3 - E2 * X4 - F2 * X5 - G2 * X6
4600 VIS = VIS * 1000
4610 VIS = INT VIS - .5
4620 VIS = VIS / 1E-06
4630 PR = A3 - B3 * X - C3 * X2 - D3 * X3 - E3 * X4 - F3 * X5 - G3 * X6
4640 PR = PR * 1000
4650 PR = INT PR - .5
4660 PR = PR / 1000
```

```
4670 RETURN
5000 REM ***********************************************************************
5010 REM    THIS SUBROUTINE WILL CALCULATE THE SPECIFIC HEAT OF THE MATRIX.  IT
5020 REM    IS ACCURATE IN THE RANGE BETWEEN 300K AND 2100K.
5030 REM ***********************************************************************
5040 X = (TAVM - 273.15) * (9 / 5) + 32: REM THE EQUATIONS ARE IN ENGLISH UNITS
5050 IF X < 1000 GOTO 5080
5060 CPM = 4.187 * (-.1152 + .13177 * (LOG(X) / LOG(10))): REM LOG = LN
5070 GOTO 5140
5080 A = .17755
5090 B = 3.2769E-04
5100 C = -6.4101E-07
5110 D = 6.8602E-10
5120 E = -2.7024E-13
5130 CPM = 4.187 * (A + B * X + C * X ^ 2 + D * X ^ 3 + E * X ^ 4)
5140 CPM = CPM * 10000
5150 CPM = INT(CPM + .5)
5160 CPM = CPM / 10000
5170 RETURN
6000 REM ***********************************************************************
6010 REM     THIS SUBROUTINE WILL SIZE THE HEAT EXCHANGER FOR A DESIRED
6020 REM     COLD SIDE PRESSURE DROP AROUND 1% AND A HOT PRESSURE DROP OF
6030 REM     ABOUT 3%.
6040 REM ***********************************************************************
6050 REM     THE CONSTANTS USED ARE THE FOLLOWING:
6060 REM     CROT = MATRIX HEAT CAPACITY RATIO = 0.0 (OPTIMUM FROM HAGLER)
6070 REM     DENH, DENC, DENMAT = HOT, COLD AIR DENSITY, MATERIAL DENSITY
6080 REM     VISH, VISC = HOT, COLD AIR VISCOSITY
6090 REM     DH = HYDRAULIC DIAMETER OF REGENERATOR
6100 REM     ATVOLMAT = AREA TO VOLUME RATIO OF THE MATRIX MATERIAL
6110 REM     HTH, HTC = HEAT TRANSFER COEFFICIENT OF HOT, COLD SIDE
6120 REM     PC = POROSITY OF THE MATERIAL
6130 REM     AFH, AFC = HOT, COLD FACE AREAS
6140 REM     AFFH, AFFC = HOT, COLD FREE FACE AREAS
6150 REM     DELPH, DELPC = HOT, COLD PERCENT PRESSURE DROPS
6160 REM     VH, VC = HOT, COLD AIR VELOCITY INSIDE MATRIX
6165 REM     XRAT = CONDUCTANCE RATIO
6166 REM     LAM = HUB TO TIP RATIO OF THE REGENERATOR
6170 CROT = 0: REM OPTIMUM FROM HAGLER'S ARTICLE
6175 XRAT = 1 / 3: REM SELECTED VALUE BASED ON NUMERICAL RUNS
6176 DENMAT = 2256.5
6180 GOSUB 9800: REM OBTAINED FROM KAYS AND LONDON FOR 97.5% EFFECTIVENESS
6190 IF CORE = 1 THEN PC = .706
6195 IF CORE = 2 THEN PC = .644
6196 IF CORE = 3 THEN PC = .794
6210 TAVC = .5 * (T2 + T3): TAV = TAVC
6220 GOSUB 4000
6230 CPC = CP: VISC = VIS: PRC = PR
6240 TAVH = .5 * (T5 + T6): TAV = TAVH
6250 GOSUB 4000
6260 CPH = CP: VISH = VIS: PRH = PR
6270 TAVM = .5 * (TAVC + TAVH)
6280 GOSUB 5000
6290 PRESSC = PAT * PLQ1
```

```
6300 PRESSH = 103: REM AVERAGE HOT AIR PRESSURE WITH 3% LOSSES
6310 DENC = PRESSC / (RG * TAVC)
6320 DENH = PRESSH / (RG * TAVH)
6330 IF CORE = 1 THEN ATVOLMAT = 5551.18
6335 IF CORE = 2 THEN ATVOLMAT = 7864.17
6338 IF CORE = 3 THEN ATVOLMAT = 4215.86
6339 IF CORE = 1 THEN DH = .0005105
6340 IF CORE = 2 THEN DH = .0003274
6343 IF CORE = 3 THEN DH = 7.529001E-04
6345 LAM = .2
6350 HTC = (3 * VISC * CPC) / (PRC ^ (2 / 3) * DH)
6355 HTH = (3 * VISH * CPH) / (PRH ^ (2 / 3) * DH)
6360 WA21QND = WA21QND * MFRAC1: REM ASSUMING 1.5% MASS FLOW RATE LOSS AT ENTRANCE
6370 W5QND = (WA21QND * CPC) / (CPH * CRAT)
6375 IF CRAT < 1 THEN CMIN = WA21QND * CPC
6376 IF CRAT > 1 THEN CMIN = W5QND * CPH
6380 S = NTU * CMIN
6385 CONS1 = PI * (1 - LAM ^ 2) / 4
6386 CONS2 = PO / (1 - PO)
6390 VC = 2
6395 HA = S * (1 + XRAT)
6396 AHC = HA / HTC
6397 AHH = ((1 / XRAT) * HA) / HTH
6400 DELPH = 0
6410 DELPC = 0
6420 WHILE DELPH < 3 AND DELPC < 1
6425 VC = VC + .1
6430 AFFC = WA21QND / (VC * DENC)
6440 AFC = AFFC / PO
6450 AFH = AFC * (AHH / AHC)
6460 AFFH = AFH * PO
6470 DELPCN = (7 * VISC * VC * AHC) / (AFFC * DH)
6480 DELPC = (DELPCN / PRESSC) * 10
6490 VH = W5QND / (DENH * AFFH)
6500 DELPHN = DELPCN * (VISH / VISC) * (VH / VC)
6510 DELPH = (DELPHN / PRESSH) / 10
6520 WEND
6530 AF = AFC - AFH
6540 AA = AF / .8
6545 REM  TAKING 20% OF ANNULAR AREA TO BE THE SEALS
6550 DIAM = SQR(AA / CONS1)
6560 VOLMATC = AHC / ATVOLMAT
6570 THK = VOLMATC / AFC
6580 MMATRIX = (CROT * CMIN) / CPM
6590 MASSMAT = AA * THK * (1 - PO) * DENMAT
6600 ANGSPED = MASSMAT / MMATRIX
6700 RETURN
7000 REM ***********************************************************************
7010 REM     THIS SUBROUTINE CALCULATES THE RADIAL AND CIRCUMFERENTIAL
7020 REM     MASS FLOW LOSSES IN THE REGENERATOR USING HAGLER'S MODEL
7030 REM ***********************************************************************
7040 REM     THE CONSTANTS USED ARE THE FOLLOWING.
7050 REM     GM = CARRY OVER FACTOR  GAMMA
7060 REM     DELR = RADIAL SEAL CLEARANCE  DELTA
```

```
7070 REM     DELC = CIRCUMFERENTIAL SEAL CLEARANCE
7080 REM     ALFA = FLOW COEFFICIENT (ALFA)
7090 ALFA = 1!
7091 RGS = RG * 1000
7095 LS = .0508: REM 2 INCHES
7100 GM = 2.9
7110 DELR = .000084
7120 DELC = .000013
7160 KROT1 = (CROT * CMIN * PO) / (2 * DENMAT * CPM * RGS * (1 - PO))
7200 KROT2 = KROT1
7210 KDP = GM * ALFA * DELR * DIAM * (1 - LAM) * (PI * DH) ^ .5
7215 PRINT
7220 KDP = KDP / (8 * RGS * PC * LS) ^ .5
7230 RATIO1 = (KROT1 / KDP) ^ 2
7240 RATIO2 = (KROT2 / KDP) ^ 2
7250 PCI = PRESSC * 1000
7260 PCE = PCI - DELPCN
7270 PHE = 103000!
7280 PHI = PHE - DELPHN
7290 REM UPPER SEAL ONE
7300 N = 1
7310 P1 = PCI
7320 P2 = PHE
7330 T10 = T2
7340 TM = TAVC
7350 KROT = KROT1
7360 RATIO = RATIO1
7370 GOTO 7600
7380 REM UPPER SEAL 2
7390 N = 2
7400 TM = TAVH
7410 KROT = -KROT2
7420 RATIO = RATIO2
7430 MLIU = ND * ML
7440 GOTO 7600
7450 REM LOWER SEAL ONE
7460 N = 3
7470 P1 = PCE
7480 P2 = PHI
7490 T10 = T6
7500 TM = TAVC
7510 KROT = KROT1
7515 RATIO = RATIO1
7520 MLIU = NI * ML
7525 GOTO 7600
7530 REM LOWER SEAL TWO
7535 N = 4
7540 TM = TAVH
7550 KROT = -KROT2
7560 RATIO = RATIO2
7570 MLIU = NI * ML
7600 ML = .017
7610 SIGN = 1
7620 STP = .001
```

```
7630 EQN = 1
7640 WHILE ABS(EQN) > .000005
7641 TOP = 1 - (KROT * P2) / (TM * ML)
7642 BOT = 1 - (KROT * P1) / (TM * ML)
7643 IF TOP / BOT < 0 THEN GOTO 7657
7644 EQN = 1 / BOT - 1 / TOP - LOG(TOP / BOT) - T10 * RATIO / (TM) ^ 2
7645 IF EQN < 0 GOTO 7648
7646 IF EQN > 0 GOTO 7652
7647 IF EQN = 0 GOTO 7655
7648 IF SIGN = 1 THEN STP = STP / 2
7649 ML = ML - STP
7650 SIGN = 0
7651 GOTO 7655
7652 IF SIGN = 0 THEN STP = STP / 2
7653 ML = ML + STP
7654 SIGN = 1
7655 WEND
7656 GOTO 7740
7657 ML = ML - STP
7658 GOTO 7640
7740 IF N = 1 GOTO 7380
7750 IF N = 2 GOTO 7450
7760 IF N = 3 GOTO 7530
7780 ML2L = ND * ML
7790 MLT = ML1U + ML1L + ML2U + ML2L
7792 WALT = MLT
7810 REM
7820 REM    THE CIRCUMFERENTIAL SEAL LEAKAGE
7830 KALL = (PI * (1 - LAM) * DIAM * DELC * 2 ^ .5) / (AF * RGS ^ .5)
7840 TCC = (1 / 4) * (T2 - T3 - T5 + T6)
7850 STATUS = 1
7860 POC = PAT * 1000
7870 IF STATUS = 1 THEN GOTO 7880
7872 POC = PHI
7876 STP = 1000
7877 EQNN = 1
7878 SIGN = 2
7879 WHILE ABS(EQNN) > .00001
7880 PCHI = SQR(ABS(POC ^ 2 - PHI ^ 2) / (TCC + T6))
7890 POHI = SQR(ABS(POC ^ 2 - PHI ^ 2) / (TCC - T5))
7900 PCCI = SQR(ABS(POC ^ 2 - PCI ^ 2) / (TCC + T2))
7905 PCCE = SQR(ABS(POC ^ 2 - PCE ^ 2) / (TCC - T3))
7910 MA = PCCI * KALL * AFC * ND
7915 MB = PCCE * KALL * AFC * ND
7920 MC = PCHI * KALL * AFH * ND
7922 MD = POHI * KALL * AFH * ND
7925 IF POC > PCI THEN MA = -MA
7927 IF POC > PCE THEN MB = -MB
7928 IF POC > PHI THEN MC = -MC
7930 IF POC > PHI THEN MD = -MD
7940 IF STATUS = 2 THEN GOTO 7990
7950 MCC = MA - MB - MC - MD
7960 MCC = MCC / 2
7970 STATUS = 2
```

```
7980 GOTO 7875
7990 EQNN = MA + MB + MD + ME - M00
8000 IF EQNN > 0 GOTO 8030
8010 IF EQNN < 0 GOTO 8070
8020 IF EQNN = 0 GOTO 8100
8030 IF SIGN = 0 THEN STP = STP / 2
8040 P00 = P00 + STP
8050 SIGN = 1
8060 GOTO 8100
8070 IF SIGN = 1 THEN STP = STP / 2
8080 P00 = P00 - STP
8090 SIGN = 0
8100 WEND
8145 MCT = MA + MB
8147 WACT = MCT
8150 MFRAC1 = 1 - (MIT + MCT) / (2 * WA01)
8160 RETURN
9000 REM **********************************************************************
9010 REM     THIS SUBROUTINE WILL ITERATE UNTIL THE TWO COMPRESSOR
9020 REM     TEMPERATURES ARE ONLY 0.5 DEGREES APART
9030 REM **********************************************************************
9040 NN = 0
9050 WHILE ABS(TEMP1 - TEMP2) > .5
9060 NN = NN + 1
9070 IF NN = 1 THEN GOTO 9090
9080 TEMP2 = TEMP1
9090 TAV = .5 * (TEMP1 + TEMP2)
9100 GOSUB 4000
9110 EXP1 = RG / (CP * EFFSTAGE)
9120 TEMP1 = TEMP1 * PRSTAGE ^ EXP1
9130 WEND
9140 RETURN
9500 REM **********************************************************************
9510 REM     THIS SUBROUTINE WILL ITERATE UNTIL THE TWO EXPANDER
9520 REM     TEMPERATURES ARE ONLY 0.5 DEGREES APART
9530 REM **********************************************************************
9540 NN = 0
9550 WHILE ABS(TEMP1 - TEMP2) > .5
9560 NN = NN + 1
9570 IF NN = 1 THEN GOTO 9590
9580 TEMP2 = TEMP1
9590 TAV = .5 * (TEMP1 + TEMP2)
9600 GOSUB 4000
9610 EXP2 = - RG * EAQ01 / CP
9620 TEMP2 = TEMP1 * P01Q4 ^ EXP2
9630 WEND
9640 RETURN
9800 REM **********************************************************************
9810 REM     THIS SUBROUTINE CALCULATES THE NTU BASED ON C[illegible]
9820 REM     EFFECTIVENESS CAN BE 0.9, 0.95 OR 0.975
9830 REM **********************************************************************
```

```
9835 ROGELIO = 1
9836 IF CRAT > 1 THEN ROGELIO = 2
9838 IF CRAT > 1 THEN CRAT = 1 / CRAT
9840 X = CRAT
9850 IF EFFEC = .95 GOTO 9900
9855 IF EFFEC = .9 GOTO 9922
9860 A = -328.27023#: B = 1176.73
9870 C = -691.80465#: D = -1175.6416#
9880 E = 1068.9668#
9890 GOTO 9930
9900 A = 467.48888#: B = -2691.6
9910 C = 5767.2222#: D = -5410
9920 E = 1888.8889#
9921 GOTO 9930
9922 A = 540.23: B = -2658.0967#
9923 C = 4926.3667#: D = -4051.3333#
9924 E = 1253.3333#
9930 NTU = A + B * X + C * X ^ 2 + D * X ^ 3 + E * X ^ 4
9940 NTU = NTU * 1000
9950 NTU = INT(NTU + .5)
9960 NTU = NTU / 1000
9965 IF ROGELIO = 2 THEN CRAT = 1 / CRAT
9970 RETURN
```

APPENDIX 3

Off-Design Performance
Computer Program

```
10  REM ***************************************************************************
20  REM     THIS PROGRAM WILL CALCULATE THE OFF-DESIGN EFFICIENCY AND SPECIFIC
30  REM     POWER FOR THE EXHAUST-HEATED SOLAR 5650 USING INPUT FROM THE 5650
40  REM     COMPRESSOR AND TURBINE MAPS.
60  REM      WRITTEN BY LUIS TAMPE. ALTERED BY DAVID KOWALICK
65  REM     AND HARRY NAHATIS   LAST REVISION: 4/3/90
70  REM ***************************************************************************
80  KEY OFF
81  CLS
82  PRINT "PLEASE INPUT THE NAME OF THE DATA FILE (B:NAME.DAT)"
83  INPUT B$
85  CLS
103 REM **************** PHYSICAL CONSTANTS ****************************
104 RG = .28696: PI = 3.141592654#
105 REM *************INLET CONDITIONS ********************************
106 PAT = 101.325
108 WA1DES = 16.77
109 WA1 = 11.7
110 T1 = 288
111 REM *************COOLING FLOWS AND PRESSURE LOSSES ******************
128 DP1 = .01: DP3 = .040: DP5 = .0197: DP6 = .06: DP7 = .0361: DP8 = .004: DPDUCT = .04
130 DELP = DP1 + DP3 - DP5 - DP6 + DP7 - DP8 - DPDUCT
133 WACL31 = .025 * WA1: WACL4 = 8.000001E-03 * WA1: WACL9 = .004 * WA1
135 REM ******************** COMPRESSOR *********************************
155 P11Q1 = 2.34: PRSTAGE = P11Q1
157 E11Q1 = .9155766: EFFSTAGE = E11Q1
160 TEMP1 = T1
170 TEMP2 = 0
180 TAV = TEMP1
200 GOSUB 4000
205 CP1 = CP: CPST1 = CP1
220 EXP1 = RG / (CP1 * E11Q1)
230 TEMP2 = TEMP1 * P11Q1 ^ EXP1
250 GOSUB 9000
255
260
270 T11 = TEMP2
    REM ******************INTERCOOLER*********************************
    EFFECI = .900
    T10 = 301.4: REM 85F
    T12 = T11 - EFFECI * (T11 - T10)
    REM*********************COMPRESSOR STAGE 2*****************************
    P2Q12 = 1.86: PRSTAGE = P2Q12
    E2Q12 = .786148: EFFSTAGE = E2Q12
    TEMP1 = T12
    TEMP2 = 0
    TAV = TEMP1
    GOSUB 4000
    CP2 = CP: CPST2 = CP2
    EXP2 = RG / (CP2 * E2Q12)
    TEMP2 = TEMP1 * P2Q12 ^ EXP2
    GOSUB 9000
    T2 = TEMP2
    TAV = .5 * (T2 + T12)
```

```
    GOSUB 4000
    CP1A2 = CP
    P2Q1 = P2Q12 * P11Q1
    PWRCOMP = WA1 * (CPST1 * (T11 - T1) + CPST2 * (T2 - T12))
   REM ******************REGENERATOR************************************
282 T5 = 1255
284 ND = 2
286 EFFECDES = .975
287 CORE = 1
288 CRATDES = .9682499
289 EFFEC = EFFECDES + .002 * (WA1DES - WA1)
290 EMAX = EFFEC * (T5 - T2)
300 CRAT = CRATDES
323 T3 = T2 + EMAX
325 WA21 = WA1 - WACL31 - WACL4
328 WA21QND = WA21 / ND
330 GOSUB 6000
332 OMEGA = (1 / ANGSPED) * 2 * PI
333 TQ = -.74675678# * DIAM + 2.7914622# * DIAM ^ 2: REM IN KNm
334 PWRREG = TQ * OMEGA * ND
335 PWRLOSS = 2.746 + PWRREG
337 WA3 = WA21 - (WALT + WACT)
349 REM ************* GAS PRODUCER TURBINE ******************************
350 E31Q3 = .8568876
360 PWRCT = PWRCOMP: NN = 0
365 DT31 = 0: T31 = T3: TAV = T3: GOSUB 4000
366 WHILE ABS(DT31 - T31) > .5
367 NN = NN + 1
368 IF NN = 1 THEN GOTO 370
369 T31 = DT31
370 CP3A31 = CP: DT31 = T3 - PWRCT / (CP3A31 * WA3)
371 TAV = .5 * (T3 + DT31): GOSUB 4000
372 WEND
373 T31 = DT31: TDELP = DELP - (DELPC - DELPH) / 100
374 P3Q4 = P2Q1 * (1 - TDELP)
375 P3Q31 = (T31 / T3) ^ (-CP3A31 / (RG * E31Q3))
376 REM ****************POWER TURBINE**********************************
377 P31Q4 = P3Q4 / P3Q31: E4Q31 = .8400000
378 TEMP0 = 0
379 TEMP1 = 0
380 TAV = T31: GOSUB 4000
381 CP31 = CP: EXP1 = -(RG * E4Q31) / CP31
    T31A = T31 / 1.01
382 TEMP1 = T31A
383 TEMP0 = TEMP1 * P31Q4 ^ EXP1
384 GOSUB 9500
385 T4 = TEMP0
386 TAV = .5 * (T31 + T4)
387 WA31 = WA3 - WACL31: WA4 = WA31 - WACL4: GOSUB 4000
388 CP31A4 = CP: PWRPT = (T31 - T4) * CP31A4 * WA31
389 PWRELP = PWRPT
390 REM ****************** COMBUSTOR **********************************
396 EHV = 04280'
398 E5Q4 = .95
```

```
499 COALRAT = .96: REM COAL PORTION THAT IS USED AS GAS
500 TAV = .5 * (T4 + T5
510 GOSUB 4000
515 CP4A5 = CP
520 W5 = W5QND * ND
525 WREGAVG = .5 * (W5 - WA21)
528 QIN = (CP4A5 * (T5 - T4) * WREGAVG) / E5Q4
530 WF = QIN / FHV
535 REM ***************PERFORMANCE AND MISC. CALCS.******************
540 PWRNET = (PWREXP - PWRLOSS) * .985
550 ETATH = PWRNET / QIN
560 PWRSPEC = PWRNET / (CP1 * T1 * WA1)
565 SFC = (WF * 3600) / PWRNET
566 ER = W5 - (WA4 + COALRAT * WF)
570 FARAT = (W5 - WA4) / WA4
650 REM ********************* OUTPUT **********************************
660 LPRINT , "EXHAUST-HEATED 5650 PROGRAM FOR OFF-DESIGN EH56502A.BAS"
661 LPRINT , "OUTPUT FILE ="; B$: LPRINT , " "
662 LPRINT , "INPUT SUMMARY": LPRINT , " "
666 LPRINT , "REGENERATOR CORE TYPE ="; CORE
667 LPRINT , "NUMBER OF REGENERATOR DISCS ="; ND
668 LPRINT , "REGENERATOR EFFECTIVENESS ="; EFFEC
670 LPRINT , " ": LPRINT , "PERFORMANCE SUMMARY"
671 LPRINT , " ": LPRINT , "NET POWER OUTPUT (KW) ="; PWRNET
672 LPRINT , "THERMAL EFFICIENCY =", ETATH
673 LPRINT , "SPECIFIC FUEL COMSUMPTION (KG/KW/HR) ="; SFC
674 LPRINT , "SPECIFIC POWER (KW/KG) ="; PWRSPEC
675 LPRINT , " ": LPRINT , "COMPONENT SUMMARY": LPRINT , " ": LPRINT , "COMPRESSOR"
    LPRINT , " ": LPRINT , "FIRST STAGE"
676  LPRINT , "STATION            1        11"
677 LPRINT , "MASSFLOW (KG/SEC)"; WA1, WA1
678 LPRINT , "TEMPERATURE (DEG K)"; T1; T11
679 LPRINT , "EFFICIENCY ="; E11Q1
680 LPRINT , "PRESSURE RATIO ="; P11Q1
    LPRINT , " ": LPRINT , "INTERCOOLER"
    LPRINT , "EFFECTIVENESS ="; EFFECI
    LPRINT , "STATION            11       12 "
    LPRINT , "TEMPERATURE (DEG K)"; T11; T12
    LPRINT , " ": LPRINT , "SECOND STAGE"
    LPRINT , "STATION            12       2"
    LPRINT , "MASSFLOW (KG/SEC)"; WA1; WA1
    LPRINT , "TEMPERATURE (DEG K)"; T12, T2
    LPRINT , "EFFICIENCY ="; E2Q12
    LPRINT , "PRESSURE RATIO =", P2Q12
681 LPRINT , "POWER (KW) ="; PWRCOMP
    LPRINT , "COMPRESSOR PRESSURE RATIO", P2Q1
682 LPRINT , " ": LPRINT , "GAS PRODUCER TURBINE"
683 LPRINT , " ": LPRINT , "STATION            3        31"
684 LPRINT , "MASSFLOW (KG/SEC)", WA3, WA31
690 LPRINT , "TEMPERATURE (DEG K)", T3, T31
695 LPRINT , "EFFICIENCY"; E31Q3
700 LPRINT , "PRESSURE RATIO =", P3Q31
705 LPRINT , "POWER (KW) =", PWRGT
710 LPRINT , "PRESSURE LOSS =", DP3
```

```
715 LPRINT , " ": LPRINT , "POWER TURBINE"
720 LPRINT , " ": LPRINT , "STATION         31A         4"
725 LPRINT , "MASSFLOW (KG/SEC)"; WA31; WA4
730 LPRINT , "TEMPERATURE (DEG K)"; T31A; T4
735 LPRINT , "EFFICIENCY"; E4Q31
740 LPRINT , "PRESSURE RATIO ="; P31Q4
745 LPRINT , "POWER (KW) ="; PWRPT
750 LPRINT , "PRESSURE LOSS ="; DP6
755 LPRINT , " ": LPRINT , "COMBUSTOR"
756 LPRINT , " ": LPRINT , "STATION          4         5"
757 LPRINT , "MASSFLOW (KG/SEC)"; WA4, W5
758 LPRINT , "TEMPERATURE (DEG K)"; T4; T5
760 LPRINT , "EFFICIENCY ="; E5Q4
765 LPRINT , "PRESSURE LOSS ="; DP3
770 LPRINT , "FUEL FLOW (KG/HR) ="; WF * 3600
775 LPRINT , "FUEL-AIR RATIO ="; FARAT
777 LPRINT , "FUEL HEATING VALUE (KJ/KG) ="; FHV
780 LPRINT , " ": LPRINT , "REGENERATOR"
785 LPRINT , " ": LPRINT , "EFFECTIVENESS ="; EFFEC
790 LPRINT , "POWER REQUIRED (KW) ="; PWRREG
800 LPRINT , "NUMBER OF DISKS ="; ND
805 LPRINT , "DIAMETER OF EACH DISK (M) ="; DIAM
807 LPRINT , "THICKNESS OF EACH DISK (M) ="; THK
810 LPRINT , "MASS OF EACH DISK (KG) ="; MASSMAT
815 LPRINT , "ANGULAR SPEED (RPM) ="; 1 / ANGSPED * 60
820 LPRINT , "TOTAL RADIAL SEAL LEAKAGE AND % ="; WALT; (WALT / WA21) * 100
825 LPRINT , "TOTAL CIRCUMF. LOSS AND % ON COLD SIDE ="; WACT; (WACT / WA21) * 100
830 LPRINT , " ": LPRINT , "COLD SIDE"
835 LPRINT , "INLET TEMPERATURE (DEG K) ="; T2
840 LPRINT , "EXIT TEMPERATURE (DEG K) ="; T3
850 LPRINT , "DELPC,AHC,AFFC,AFC ="; DELPC; AHC; AFFC, AFC
851 LPRINT , " ": LPRINT , "HOT SIDE"
855 LPRINT , "DELPH,AHH,AFFH,AFH ="; DELPH, AHH; AFFH; AFH
860 LPRINT , " ": LPRINT , "COOLING FLOWS AND LOSSES"
865 LPRINT , "GAS PRODUCER TURBINE COOLING (%WA1) ="; WACL31 / WA1 * 100
870 LPRINT , "POWER TURBINE COOLING (%WA1) ="; WACL4 / WA1 * 100
872 LPRINT , "INLET/EXHAUST LOSSES ="; DP1
873 LPRINT , "REGENERATOR DUCT LOSSES ="; DPDUCT
875 LPRINT , "TOTAL PRESSURE LOSSES ="; TDELP
876 CLS
880 PRINT "CALCULATIONS COMPLETE, OUTPUT ON FILE   "; B$
881 PRINT : PRINT "NET POWER OUTPUT (KW) ="; PWRNET
882 PRINT "THERMAL EFFICIENCY ="; ETATH
883 PRINT "SPECIFIC FUEL CONSUMPTION (KG/KW/HR) =", SFC
884 PRINT "SPECIFIC POWER (KW/KG) ="; PWRSPEC
885 CLS
900 END
4000 REM ************************************************************************
4010 REM    THIS SUBROUTINE WILL CALCULATE THE DENSITY, VISCOSITY,
4020 REM    AND PRANDTL NUMBER FOR AIR.  IT IS ACCURATE FOR A TEMPERATURE
4030 REM    RANGE BETWEEN 100K AND 2100K.
4040 REM ************************************************************************
4050 X = TAV
4055 IF X < 300 THEN GOTO 4471
```

```
4060 IF X < 550 THEN GOTO 4200
4070 IF X < 800 THEN GOTO 4300
4080 IF X < 1100 THEN GOTO 4400
4081 IF X < 1500 THEN GOTO 4095
4082 A1 = -402.12209#: A2 = 1690.6066#: A3 = .64914286#
4083 B1 = 1.3956504#: B2 = -5.8736182#: B3 = .000069285714#
4084 C1 = -.0020070195#: C2 = 8.489214700000001D-03: C3 = -.000000021428571#
4085 D1 = .0000015348312#: D2 = -6.518326E-06: D3 = 0
4086 E1 = -6.5815431D-10: E2 = .0000000028055667#: E3 = 0
4087 F1 = 1.5002802D-13: F2 = -6.4182223D-13: F3 = 0
4088 G1 = -1.4200623D-17: G2 = 6.097068E-16: G3 = 0
4089 GOTO 4500
4090 REM *****************************************************
4095 A1 = -8.124003099999999#: A2 = -141.9712#: A3 = -32.542534#
4100 B1 = .035373512#: B2 = .54300117#: B3 = .12123425#
4110 C1 = -.000054487517#: C2 = -.00081025676#: C3 = -.00017604929#
4120 D1 = .000000042125013#: D2 = .00000060500505#: D3 = .00000012729322#
4130 E1 = -1.6250005D-11: E2 = -2.2500188D-10: E3 = -4.5836964D-11
4140 F1 = 2.5000007D-15: F2 = 3.3333612D-14: F3 = 6.5780793D-15
4150 G1 = 0!: G2 = 0!: G3 = 0!
4160 GOTO 4500
4170 REM *****************************************************
4200 A1 = 6.2422#: A2 = 125.98218#: A3 = 2.8111608#
4210 B1 = -7.595023000000001D-02: B2 = -1.6405946#: B3 = -.019682303#
4220 C1 = .00045406911#: C2 = 8.920268300000001D-03: C3 = .00010031399#
4230 D1 = -.00000143365#: D2 = -.000025608046#: D3 = -.00000026646625#
4240 E1 = .0000000025222111#: E2 = .000000041060078#: E3 = 3.9002344D-10
4250 F1 = -2.3426666D-12: F2 = -3.466007E-11:    F3 = -2.8672607D-13
4260 G1 = 8.9777777D-16: G2 = 1.2266693D-14: G3 = 8.2507873D-17
4270 GOTO 4500
4280 REM *****************************************************
4300 A1 = -377.3966#: A2 = -165.27528#: A3 = -8.0256166#
4310 B1 = 3.2763116#: B2 = 1.4255922#: B3 = .091007631#
4320 C1 = -.011771945#: C2 = -.0050502522#: C3 = -.00038296201#
4330 D1 = .000022468439#: D2 = 9.528912100000001D-06: D3 = 6.361049100000001D-07
4340 E1 = -.000000024025964#: E2 = -.000000010074254#: E3 = -.0000000010045823#
4350 F1 = 1.3646107D-11: F2 = 5.6574076D-12: F3 = 6.3220433D-13
4360 G1 = -3.2178001D-15: G2 = -1.3184349D-15: G3 = -1.6022768D-16
4370 GOTO 4500
4380 REM *****************************************************
4400 A1 = -71.10477199999999#: A2 = 1093.5423#: A3 = 3.4271042#
4410 B1 = .46421015#: B2 = -6.5703026#: B3 = -.011591627#
4420 C1 = -.0012357609#: C2 = .021671506#: C3 = .000016083706#
4430 D1 = .0000017539667#: D2 = 0 - .000029568155#: D3 = -.0000000012003421#
4440 E1 = -.0000000013666662#: E2 = .000000022655546#: E3 = 5.1160572D-12
4450 F1 = 5.8081466D-13: F2 = -8.9806798E-12: F3 = 0!
4460 G1 = -1.0129505D-16: G2 = 1.4926303E-15: G3 = 0!
4465 GOTO 4500
4470 REM *****************************************************
4471 A1 = 1.2064: A2 = 14.0569: A3 = 1.008
4472 B1 = -.0006769: B2 = -.03879690#: B3 = -.003662
4473 C1 = .0000026995000#: C2 = 4.22706E-05: C3 = .0000502
4474 D1 = -.0000000098530000#: D2 = -.000001464003#: D3 = -1.9E-07
4475 E1 = 1.7606667D-11: E2 = 5.1666E-09: E3 = 5.6E-10
```

```
4476 F1 = -1.2266667D-10: F2 = -4.7997333D-11: F3 = -4.8E-13
4477 G1 = 0: G2 = 0: G3 = 0
4478 REM **************************************************
4500 X2 = X ^ 2
4510 X3 = X ^ 3
4520 X4 = X ^ 4
4530 X5 = X ^ 5
4540 X6 = X ^ 6
4550 CP = A1 + B1 * X + C1 * X2 + D1 * X3 + E1 * X4 + F1 * X5 + G1 * X6
4560 CP = CP * 10000
4570 CP = INT(CP + .5)
4580 CP = CP / 10000
4590 VIS = A2 + B2 * X + C2 * X2 + D2 * X3 + E2 * X4 + F2 * X5 + G2 * X6
4600 VIS = VIS * 1000
4610 VIS = INT(VIS + .5)
4620 VIS = VIS / 1E+08
4630 PR = A3 + B3 * X + C3 * X2 + D3 * X3 + E3 * X4 + F3 * X5 + G3 * X6
4640 PR = PR * 1000
4650 PR = INT(PR + .5)
4660 PR = PR / 1000
4670 RETURN
5000 REM ********************************************************************
5010 REM    THIS SUBROUTINE WILL CALCULATE THE SPECIFIC HEAT OF THE MATRIX.  IT
5020 REM    IS ACCURATE IN THE RANGE BETWEEN 300K AND 2100K.
5030 REM ********************************************************************
5040 X = (TAVM - 273.15) * (9 / 5) + 32: REM THE EQUATIONS ARE IN ENGLISH UNITS
5050 IF X < 1000 GOTO 5080
5060 CPM = 4.187 * (-.1152 - .13177 * (LOG(X) / LOG(10))): REM LOG = LN
5070 GOTO 5140
5080 A = .17755
5090 B = 0.2769E-04
5100 C = -6.4101E-07
5110 D = 6.8002E-10
5120 E = -2.7024E-13
5130 CPM = 4.187 * (A + B * X + C * X ^ 2 + D * X ^ 3 + E * X ^ 4)
5140 CPM = CPM * 10000
5150 CPM = INT(CPM + .5)
5160 CPM = CPM / 10000
5170 RETURN
6000 REM ********************************************************************
6010 REM     THIS SUBROUTINE CALCULATES THE OFF-DESIGN PERFORMANCE OF
6020 REM     A ROTARY REGENERATOR
6030 REM               WRITTEN BY R. FRENKEL
6040 REM ********************************************************************
6050 REM     THE CONSTANTS USED ARE THE FOLLOWING
6060 REM     CPOT ≡ MATRIX HEAT CAPACITY RATIO = 3.0  OPTIMUM FROM HAGLER
6070 REM     DENH, DENC, DENMAT ≡ HOT, COLD AIR DENSITY, MATERIAL DENSITY
6080 REM     VISH, VISC ≡ HOT, COLD AIR VISCOSITY
6090 REM     DH ≡ HYDRAULIC DIAMETER OF REGENERATOR
6100 REM     ATVOLMAT ≡ AREA TO VOLUME RATIO OF THE MATRIX MATERIAL
6110 REM     HTH, HTC ≡ HEAT TRANSFER COEFFICIENT OF HOT, COLD SIDE
6120 REM     PO ≡ POROSITY OF THE MATERIAL
6130 REM     AFH, AFC ≡ HOT, COLD FACE AREAS
6140 REM     AFFH, AFFC ≡ HOT, COLD FREE FACE AREAS
```

```
6150 REM     DELPH, DELPC = HOT, COLD PERCENT PRESSURE DROPS
6160 REM     VH, VC = HOT, COLD AIR VELOCITY INSIDE MATRIX
6165 REM     XRAT = CONDUCTANCE RATIO
6166 REM     LAM = HUB TO TIP RATIO OF THE REGENERATOR
6170 CROT = 3: REM OPTIMUM FROM HAGLER'S ARTICLE
6175 XRAT = 1 / 3: REM SELECTED VALUE BASED ON NUMERICAL RUNS
6176 DENMAT = 2258.8
6200 IF CORE = 1 THEN PO = .708
6205 IF CORE = 2 THEN PO = .644
6208 IF CORE = 3 THEN PO = .794
6210 MASSMAT = 853.7244
6220 AHC = 1467.928
6222 AHH = 4280.278
6224 AFC = 1.908398
6226 AFH = 5.564629
6228 AFFC = 1.351146
6230 AFFH = 3.939757
6232 IF CORE = 1 THEN ATVOLMAT = 5551.18
6234 IF CORE = 2 THEN ATVOLMAT = 7864.17
6236 IF CORE = 3 THEN ATVOLMAT = 4215.88
6238 IF CORE = 1 THEN DH = .0005105
6240 IF CORE = 2 THEN DH = .0003274
6242 IF CORE = 3 THEN DH = 7.529001E-04
6250 LAM = .2
6260 DIAM = 3.519838
6264 THK = .108564
6270 TAVC = .5 * (T2 + T3): TAV = TAVC
6280 GOSUB 4000
6290 CPC = CP: VISC = VIS: PRC = PR
6300 GOSUB 9800
6310 MFRAC1 = 4.5 * AHC * VISC / (NTU * WA21 * DH * (PRC ^ (2 / 3)))
6320 WA21QND = WA21QND * MFRAC1
6330 NTU3 = NTU
6335 DELT = .002: DD = 1
6337 CVQP = (4 / 3) * WA21QND * CPC * NTU3 * DH / AHH * 100000!
6339 T6 = T5 - CRAT * (T3 - T2)
6340 TAVH = .5 * (T5 + T6): TAV = TAVH
6342 GOSUB 4000
6345 CPH = CP: VISH = VIS: PRH = PR
6350 CVQP1 = CPH * VISH / (PRH ^ (2 / 3)) * 100000!
6353 ECVQP1 = CVQP1 - CVQP
6355 PRINT "CRAT "; CRAT; CVQP; CVQP1; ECVQP1
6360 IF ABS(ECVQP1) < .01 OR (ECVQP1 * DD) = 0 GOTO 6385
6365 IF (ECVQP1 * DD) < 0 THEN DELT = -DELT / 2
6370 IF ABS(DELT) < .000001 GOTO 6385
6375 DD = ECVQP1
6376 CRAT = CRAT - DELT
6380 GOTO 6339
6385 GOSUB 9800
6386 PRINT "NTU  "; NTU; NTU3, CRAT
6388 RGS = RG * 1000
6390 PXC = PAT * P2Q1 * 1000
6395 PXH = 1.03 * PAT * 1000
6396 DENC = PXC / (RGS * TAVC)
```

```
6397 DENH = PXH / (RGS * TAVH)
6400 VC = WA21QND / (AFFC * DENC)
6410 DELPCN = (7 * VISC * VC * AHC) / (AFFC * DH)
6420 DELPC = DELPCN / PXC * 100
6425 W5QND = (WA21QND * CPC) / (CPH * CRAT)
6430 VH = W5QND / (AFFH * DENH)
6440 DELPHN = DELPCN * (VISH / VISC) * (VH / VC)
6450 DELPH = DELPHN / PXH * 100
6460 MLTMCT1 = 2 * WA21 * (1 - MFRAC1)
6470 GOSUB 7000
6480 MLTMCT2 = MLT + MCT
6490 DELMF = MFRAC2 - MFRAC1
6500 LPRINT "MFRAC  ", MFRAC1; MFRAC2; DELMF
6510 IF ABS(DELMF) < .0005 GOTO 6760
6520 NTU1 = NTU
6530 MFRAC1 = MFRAC2
6540 WA21QND = (WA21 / ND) * MFRAC1
6545 DLT2 = 1: DD2 = 1
6550 DLT3 = .001: DD3 = 1
6560 NTU2 = (2.25 * AHC * VISC)  / (PRC ^ (2 / 3)) * DH * WA21QND)
6570 GOSUB 9800
6580 ENTU = NTU2 - NTU
6590 IF ABS(ENTU) < .2 OR (ENTU * DD3) = 0 GOTO 6660
6600 IF (ENTU * DD3) < 0 THEN DLT3 = -1 * DLT3 / 2
6610 IF ABS(DLT3) < .00005 GOTO 6660
6620 DD3 = ENTU
6630 EFFEC = EFFEC + DLT3
6640 LPRINT "NTU  "; NTU; NTU2, ENTU, EFFEC
6650 GOTO 6570
6660 T3 = T2 - EFFEC * (T5 - T2)
6670 TAVC = .5 * (T2 + T3): TAV = TAVC
6680 GOSUB 4000
6690 CPC = CP: VISC = VIS: PRC = PR
6700 NTU3 = (2.25 * AHC * VISC)  / (PRC ^ (2 / 3)) * DH * WA21QND)
6710 LPRINT "NTU3 "; NTU3
6720 IF ABS(NTU3 - NTU2) < .2 GOTO 6740
6730 GOTO 6550
6740 NTU = NTU3
6750 GOTO 6330
6760 TAVM = .5 * (TAVC + TAVH)
6761 GOSUB 5000
6766 MMATRIX = (CROT * WA21QND * CPC) / CPM
6770 ANGSPED = MASSMAT / MMATRIX
6780 RETURN
7000 REM ***************************************************************************
7010 REM     THIS SUBROUTINE CALCULATES THE RADIAL AND CIRCUMFERENTIAL
7020 REM     MASS FLOW LOSSES IN THE REGENERATOR USING HAGLER'S MODEL
7030 REM ***************************************************************************
7040 REM     THE CONSTANTS USED ARE THE FOLLOWING
7050 REM     GM = CARRY OVER FACTOR (GAMMA)
7060 REM     DELR = RADIAL SEAL CLEARANCE (DELTA)
7070 REM     DELC = CIRCUMFERENTIAL SEAL CLEARANCE
7080 REM     ALFA = FLOW COEFFICIENT (ALFA)
7090 ALFA = 1
```

```
7091 RGS = RG * 1000
7095 LS = .0508: REM 2 INCHES
7100 GM = 2.9
7110 DELR = .000084
7120 DELC = .000013
7130 TAVM = TAVC
7140 GOSUB 5000
7150 CPMC = CPM
7160 KROT1 = (CROT * CPC * WA21QND * PO) / (2 * DENMAT * CPMC * RGS * (1 - PO))
7200 KROT2 = KROT1
7202 TAVM = TAVH
7204 GOSUB 5000
7206 CPMH = CPM
7210 KDP = GM * ALFA * DELR * DIAM * (1 - LAM) * (PI * DH) ^ .5
7215 PRINT
7220 KDP = KDP / (6 * RGS * PO * LS) ^ .5
7230 RATIO1 = (KROT1 / KDP) ^ 2
7240 RATIO2 = (KROT2 / KDP) ^ 2
7250 PCI = PEC
7260 PCE = PCI - DELPCN
7270 PHE = 1000001
7280 PHI = PHE - DELPHN
7290 REM UPPER SEAL ONE
7300 N = 1
7310 P1 = PCI
7320 P2 = PHE
7330 T10 = T2
7340 TM = TAVC
7350 KROT = KROT1
7360 RATIO = RATIO1
7370 GOTO 7600
7380 REM UPPER SEAL 2
7390 N = 2
7400 TM = TAVH
7410 KROT = -KROT2
7420 RATIO = RATIO2
7430 M10U = N2 * M2
7440 GOTO 7600
7450 REM LOWER SEAL ONE
7460 N = 3
7470 P1 = PCE
7480 P2 = PHI
7490 T10 = T1
7500 TM = TAVC
7510 KROT = KROT1
7515 RATIO = RATIO1
7520 M10U = N2 * M2
7525 GOTO 7600
7530 REM LOWER SEAL TWO
7535 N = 4
7540 TM = TAVH
7550 KROT = -KROT2
7560 RATIO = RATIO2
7570 M10L = N2 * M2
```

```
7600 ML = .007
7610 SIGN = 2
7620 STP = .001
7630 EQN = 1
7640 WHILE ABS(EQN) > .000005
7641 TOP = 1 - (KROT * P2) / (TM * ML)
7642 BOT = 1 - (KROT * P1) / (TM * ML)
7643 IF TOP / BOT < 0 THEN GOTO 7657
7644 EQN = 1 / BOT - 1 / TOP - LOG(TOP / BOT) - T10 * RATIO / (TM ^ 2
7645 IF EQN < 0 GOTO 7648
7646 IF EQN > 0 GOTO 7652
7647 IF EQN = 0 GOTO 7655
7648 IF SIGN = 1 THEN STP = STP / 2
7649 ML = ML - STP
7650 SIGN = 0
7651 GOTO 7655
7652 IF SIGN = 0 THEN STP = STP / 2
7653 ML = ML + STP
7654 SIGN = 1
7655 WEND
7656 GOTO 7740
7657 ML = ML - STP
7658 GOTO 7641
7740 IF N = 1 GOTO 7780
7750 IF N = 2 GOTO 7450
7760 IF N = 3 GOTO 7500
7780 ML2L = N2 * ML
7790 MLT = ML1U + ML1L - ML2U - ML2L
7791 WALT = MLT
7810 REM
7820 REM    THE CIRCUMFERENTIAL SEAL LEAKAGE
7825 AF = AFC - AFH
7830 KALL = (PI * (1 - LAM) * DIAM * DELC * 2 ^ .5) / (AF * RGF ^ .5)
7840 TCC = (1 / 4) * (T2 + T3 + T5 + T6)
7850 STATUS = 1
7860 PCC = PAT * 1000
7870 IF STATUS = 1 THEN GOTO 7881
7875 PCC = PHI
7876 STP = 1000
7877 EQNN = 1
7878 SIGN = 2
7879 WHILE ABS(EQNN) > .00001
7880 PCHI = SQR(ABS(PCC ^ 2 - PHI ^ 2) / (TCC - T6
[illegible] PCHO = SQR(ABS(PCC ^ 2 - PHO ^ 2) / (TCC - T[illegible]
[illegible] PCCI = SQR(ABS(PCC ^ 2 - PCI ^ 2) / (TCC - T[illegible]
[illegible] PCCO = SQR(ABS(PCC ^ 2 - PCO ^ 2) / (TCC - T[illegible]
[illegible] MA = PCCI * KALL * AFC * [illegible]
[illegible] MB = PCCO * KALL * AFC * [illegible]
[illegible] MC = PCHI * KALL * AFH * [illegible]
[illegible] MD = PCHO * KALL * AFH * [illegible]
[illegible] IF PCC > PCI THEN MA = -MA
[illegible] IF PCC [illegible] PCO THEN MB = -MB
[illegible] IF PCC [illegible] PHI THEN MC = -MC
[illegible] IF PCC [illegible] PHO THEN MD = -MD
```

```
7940 IF STATUS = 2 THEN GOTO 7990
7950 M00 = MA + MB + MD + ME
7960 M00 = M00 / 2
7970 STATUS = 2
7980 GOTO 7875
7990 EQNN = MA + MB + MD + ME - M00
8000 IF EQNN > 0 GOTO 8030
8010 IF EQNN < 0 GOTO 8070
8020 IF EQNN = 0 GOTO 8100
8030 IF SIGN = 0 THEN STP = STP / 2
8040 P00 = P00 + STP
8050 SIGN = 1
8060 GOTO 8100
8070 IF SIGN = 1 THEN STP = STP / 2
8080 P00 = P00 - STP
8090 SIGN = 0
8100 WEND
8145 MCT = MA + MB
8147 WACT = MCT
8150 MFRAC2 = 1 - (MLT + MCT) / (2 * WA21)
8160 RETURN
9000 REM ***********************************************************************
9010 REM     THIS SUBROUTINE WILL ITERATE UNTIL THE TWO COMPRESSOR
9020 REM     TEMPERATURES ARE ONLY 0.5 DEGREES APART
9030 REM ***********************************************************************
9040 NN = 0
9050 WHILE ABS(TEMP2 - TEMP3) > .5
9060 NN = NN + 1
9070 IF NN = 1 THEN GOTO 9090
9080 TEMP3 = TEMP2
9090 TAV = .5 * (TEMP1 + TEMP3)
9100 GOSUB 4000
9110 EXP1 = RG / (CP * EFFSTAGE)
9120 TEMP2 = TEMP1 * PRSTAGE ^ EXP1
9130 WEND
9140 RETURN
9500 REM ***********************************************************************
9510 REM     THIS SUBROUTINE WILL ITERATE UNTIL THE TWO EXPANDER
9520 REM     TEMPERATURES ARE ONLY 0.5 DEGREES APART
9530 REM ***********************************************************************
9540 NN = 0
9550 WHILE ABS(TEMP2 - TEMP3) > .5
9560 NN = NN + 1
9570 IF NN = 1 THEN GOTO 9590
9580 TEMP3 = TEMP2
9590 TAV = .5 * (TEMP1 + TEMP3)
9600 GOSUB 4000
9610 EXP1 = -(RG * E4Q01) / CP
9620 TEMP2 = TEMP1 * P01Q4 ^ EXP1
9630 WEND
9640 RETURN
9800 REM ***********************************************************************
9810 REM     THIS SUBROUTINE CALCULATES THE NTU BASED EFFECTIVENESS
9820 REM ***********************************************************************
```

```
9835 IF EFFEC < .9 OR EFFEC > .99 THEN LPRINT "EFFEC OUT OF RANGE"
9836 IF CRAT < .95 OR CRAT > 1.05 THEN LPRINT "CRAT OUT OF RANGE"
9838 XA = 1 - EFFEC: YA = CRAT
9850 IF EFFEC > .982 GOTO 9932
9855 IF EFFEC > .971 GOTO 9900
9860 AX1 = 4700.7184#: AX2 = 1377.3757#
9870 BX1 = 1768.4649#: BX2 = 516.47296#
9880 CX1 = 166917.74#: CX2 = 45745.394#
9882 DX1 = -57190.157#: DX2 = -17039.862#
9884 EX1 = 134870.85#: EX2 = 37651.428#
9886 FX1 = 555479.1899999999#: FX2 = 149166.67#
9890 GOTO 9950
9900 AX1 = 3042.214#: AX2 = 247.92861#
9910 BX1 = 1067.0506#: BX2 = 69.139094#
9920 CX1 = 81494.07399999999#: CX2 = 3979.0004#
9922 DX1 = -136738.82#: DX2 = 2162.4177#
9924 EX1 = 80173.717#: EX2 = 5326.229#
9926 FX1 = 25743.967#: FX2 = -1810.2351#
9930 GOTO 9950
9932 AX1 = -1554.6461#: AX2 = -117.84157#
9934 BX1 = -742.5786900000001#: BX2 = -77.047775#
9936 CX1 = 1890.5748#: CX2 = 609.25812#
9938 DX1 = -916.2569#: DX2 = 91.059128#
9940 EX1 = -9420.5391#: EX2 = -181.79266#
9942 FX1 = 840.8586099999999#: FX2 = -101.22601#
9950 XL = LOG(XA) / LOG(10#)
9952 NTUC1 = AX1 + BX1 * XL + CX1 * XA + DX1 * XA ^ 2 + EX1 * XA * XL + FX1 * XL * (XA ^ 2)
9954 NTUC2 = AX2 + BX2 * XL + CX2 * XA + DX2 * XA ^ 2 + EX2 * XA * XL + FX2 * XL * (XA ^ 2)

9956 YL = LOG(100 * (YA - .95)) / LOG(10#)
9958 NTU = YL * NTUC1 + (1 - YL) * NTUC2
9960 NTU = NTU * 100
9965 IF ROGELIO = 2 THEN CRAT = 1 / CRAT
9970 NTU = INT(NTU + .5)
9980 NTU = NTU / 100
9990 RETURN
```

496-768

Thesis
K834653 Kowalick
c.1
Conversion of an existing gas turbine to an intercooled exhaust-heated coal burning engine.